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THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

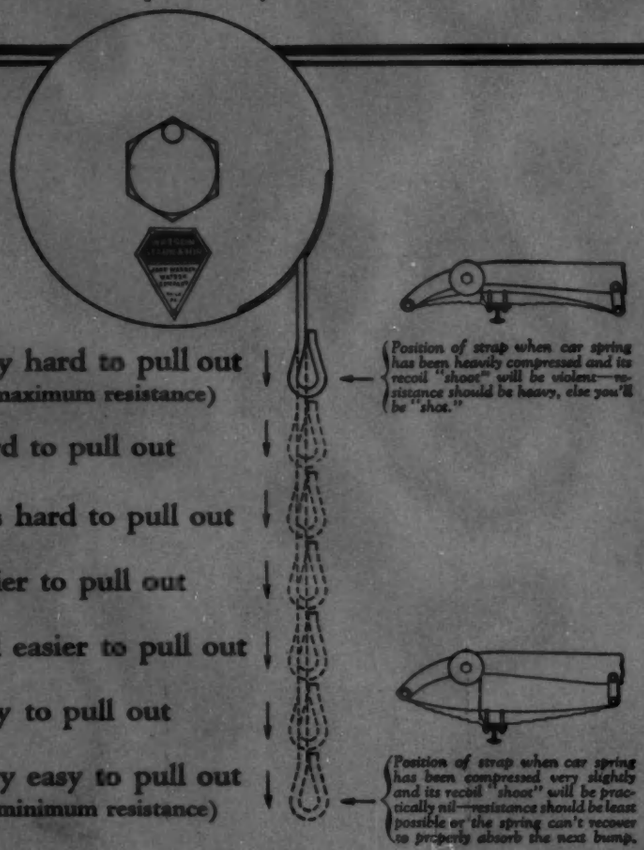
OCTOBER 1923

SOCIETY OF AUTOMOTIVE ENGINEERS INC.
29 WEST 39TH STREET NEW YORK

THE ACID TEST

The nature of the mechanism inside the device will tell whether or not it can stand the gaff, but results at the end of the strap tell whether or not the principle is right or wrong for producing easy riding

Clamp any recoil check in a vise, brace your foot up against the bench, and then tug on the strap. According to whether the strap offers less and less resistance or more and more resistance as you continue to pull it out you can tell whether the device is right or wrong.


Very hard to pull out
(maximum resistance)
Hard to pull out
Less hard to pull out
Easier to pull out
Still easier to pull out
Easy to pull out
Very easy to pull out
(minimum resistance)
Position of strap when car spring has been heavily compressed and its recoil "shoot" will be violent—resistance should be heavy, else you'll be "shot."
Position of strap when car spring has been compressed very slightly and its recoil "shoot" will be practically nil—resistance should be least possible or the spring can't recover to properly absorb the next bump.

YOU will find that a STABILATOR strap, as here illustrated, offers very stubborn resistance at the *beginning* of the pull and gradually resists less and less as you keep pulling it out. Thus a STABILATOR is not, in effect, a jump strap, but is a brake which automatically resists heaviest when the car spring is fully compressed and its recoil force is most violent—and resists less and less accordingly as the car spring is less compressed and its recoil force is *less* violent.

This resisting of each varying recoil force in *proportion* to that force is nothing more than an adherence to the same logic which tells us to put small brakes on the wheels of a Ford car and very large and powerful brakes on the wheels of a Mack 5-ton truck.

You need not take anyone's "say so" as to whether or not this, that or the other device offers correct, proportional resistance to the varying forces of spring recoil—just clamp the thing in a vise, tug on the strap, and *know* what it gives.

JOHN WARREN WATSON COMPANY, PHILADELPHIA
Twenty-fourth and Locust Streets

WATSON
STABILATORS
Change the Whole Nature of Your Car

THE

JOURNAL
OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

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H. W. ALDEN, *President* COKER F. CLARKSON, *Secretary* C. B. WHITTELSEY, *Treasurer*

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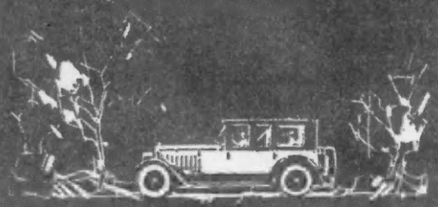

WYMAN-GORDON
The Crankshaft Makers
Worcester, Mass.
Harvey, Ill.

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

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Chronicle and Comment

The Production Meeting

LAST fall the Society held a Production Meeting at Detroit, the first of its kind held by the Society and probably the best attended professional meeting of the Society. The second Production Meeting will be held at Cleveland this month, and this meeting is drawn to your attention to show that the question of production on a quality basis is having the attention of engineers. (See page 347 this issue.) Efforts are being made to draw the production and the engineering departments into the close harmony and cooperation that is desirable if we are to serve the public honestly and to the best of our ability.—C. S. Dahlquist.

National Aeronautic Convention

AERONAUTICAL engineering progress in this Country has been rapid during the past year. Many speed, altitude and duration records of importance are now held by American airplane pilots. We have seen the wide expanse of our Continent traversed without stop. The Air Mail, in addition to maintaining its past record of superior performance, has proved its ability to fly the mail from Coast to Coast in a little more than a day. Night flying has been undertaken and proved practical. American engineers and artisans have constructed the ZR-1 and demonstrated her fitness as an air-carrier. These accomplishments are sufficient reason for pardonable pride in the proficiency of our aeronautical engineers, the more so because of the apathy displayed by the general public and the large majority of our business men.

As a fitting conclusion to this year of aviation progress, the Third National Convention of the National Aeronautic Association and International Air Races are being held in St. Louis Oct. 1-3. May these two important events stimulate a greater and more sincere interest in commercial air-transport, the all-important backbone of a prosperous aeronautical industry.

The Motorbus

THE cooperation between the American Electric Railway Association and the Society of Automotive Engineers, inaugurated by the address of President Emmons of the former organization at the Automotive Transportation meeting of the Society in Cleveland last April, will be continued at the A. E. R. A. Convention at Atlantic City this month. In the afternoon of Tuesday, Oct. 9, President Alden, of the Society, will present a discussion of a report on the subject of Bus Operation, prepared by the Transportation and Traffic Association, an affiliated section of the American Electric Railway Association. It is expected that other members of the Society will take part in the discussion of this very interesting and important subject. The traction lines are naturally giving greater and greater attention to the possibilities and proper economic status of the motorbus, which undoubtedly is a medium of transportation that will henceforth be much more widely availed of.

Martyrs to Science

DURING the afternoon of Sept. 20 an explosion of great violence occurred in the engine testing laboratory in the Bureau of Standards, City of Washington. One man was killed instantly, three others were injured so seriously that they died during the night, and four others were seriously burned or cut.

The explosion occurred in the north altitude chamber in which there was being conducted an acceleration test at low temperature of an automobile engine using aviation gasoline. The work in progress was part of a cooperative fuel investigation undertaken jointly by the Society, the American Petroleum Institute and the National Automobile Chamber of Commerce. It is stated officially and there appears to be no doubt that the explosion was due to the ignition of an explosive mixture in the chamber. No one was in the chamber at the time of the explosion, but flame issuing from it burned directly those in its path outside the chamber and ignited inflammable material including the fuel-measuring apparatus used in the test.

The dead are: Stephen M. Lee, a junior member of the Society who was in charge of the test in progress; James E. Kendig, assistant electrical engineer of the Bureau who was assisting in the test work; Urban J. Cook, machinist, and Dr. Logan L. Lauer, who was not engaged in any way in the work. Those seriously injured are Roger Birdsell, research associate engaged in the investigation, H. K. Cummings, C. W. Elliot and F. E. Richardson. The heroism of members of the Bureau

staff in rescuing the injured from the furiously burning wreckage minimized the loss of life.

Those suffering from this most deplorable disaster, the members of the families of those killed and injured, the officers of the Bureau and all those connected with the nationally important investigation, have the deepest sympathy of all of the members. As has been aptly said by Secretary of Commerce Hoover, the men were martyrs to the experimental work through which science finds its advances toward public usefulness.

Our Japanese Members

NO peace-time world calamity has stirred the sympathies of our Nation more deeply than the recent earthquake in Japan. Concrete expression of this feeling is going forward to the Japanese through the ever-faithful services of the Red Cross. However, it seems fitting to record here an expression of condolence to the members of the Society who were directly affected because of their residence in the quake and fire-swept areas. Our records show that fifteen of these resided in Tokyo. It is hoped that all were fortunate enough to survive this deplorable disaster so that their engineering knowledge and experience may be available for the labors of reconstruction.

Practical Brake Testing

ORDINANCES requiring that automobile brakes be maintained at a certain standard degree of efficiency exist in many of the larger cities, but their enforcement has been impractical for many reasons. Most important of these has been the lack of a simple and accurate method of conducting brake tests. The staff of the Bureau of Standards is offering a solution to this problem in the form of a registering decelerometer by means of which brake performance can be readily determined.

In conducting a brake test, the decelerometer is merely placed upon the floor of the car under observation. The driver is requested to attain a selected speed, apply the brakes with his full strength and keep them applied until a dead stop is effected. During the period of braking, the decelerometer registers the rate of deceleration in feet per second per second on a suitable dial. The observer can see readily whether the rate of deceleration has always exceeded a minimum considered to represent acceptable braking performance.

This Bureau of Standards method has many advantages. The tests can be made by untrained observers; the readings are accurate and not subject to correction for numerous variables; the instrument itself is portable and self-contained and requires no connection with the vehicle, engine or wheels. Tests can be made in a very short time since no measurements of distances need be taken. It is not improbable that this method of brake testing will come into universal use, particularly if the police or other agencies attempt to enforce brake-inspection ordinances.

1922 Transactions

COPIES of Part I of THE TRANSACTIONS for 1922 have been mailed to those of the members who ordered them. In accordance with the resolution of the Council, copies of THE TRANSACTIONS for the last half of 1922 will be sent without charge to members who paid dues for the first half of the fiscal year beginning Oct. 1, 1922, provided a written request is received within 60 days after a notice to order has been published. Copies of this part of THE TRANSACTIONS cannot be guaranteed to members who do not place their orders by Nov. 5. Members who have not sent in the order blank for Part II of THE TRANSACTIONS for 1922 should do so at once. The blank will be found printed in the advertising section of the September JOURNAL, p. 83.

Proper Dimensioning

THE report submitted by Earle Buckingham on Tolerances, which appears in Tentative Standardization Work in this issue of THE JOURNAL, is of exceptional value. Those who do not agree with his statement that non-mating surfaces should have bilateral or unilateral tolerances, depending on the nature of the variations which develop in manufacture, and that, with few exceptions, mating surfaces should have unilateral tolerances, should submit their views to the Society for consideration. The entire matter is too far-reaching to expect that acceptable decisions thereon can be made without the active cooperation of the industry at large.

A. E. A. Convention

AT the seventh annual summer meeting of the Automotive Electric Association at Eaglesmere Park, Pa., Sept. 11 to 14, at which joint sessions were also held with members of the Automotive Electric Service Association, a number of subjects of importance to the electrical automotive apparatus manufacturers and servicing organizations were discussed. The Standardization Committee's report on the adoption of four standard sizes of starting motors, to be referred to as Nos. 1, 2, 3 and 4, was approved by the Board of Governors. It was believed that this range of sizes, or capacities, will be adequate for at least a large proportion of automobiles electrically equipped. It was intimated that the Battery Division of the committee might consider standardizing a range of capacities for batteries to be used with the standardized sizes of starting motors and generators, the A. E. A. having previously adopted four sizes of generator. The committee also reported progress toward developing a standard method of making torque tests on starting motors and discussed the desirability of having a standard electrical test for ignition apparatus.

Several members of the committee whose investigation as to why the S. A. E. Standards for generator and starting motor mountings are not used more extensively, especially by the larger car manufacturers, explained that it is usually because of reasons important enough to the individual manufacturers; that probably most of the smaller manufacturers do use the standards, and that it would not be reasonable to expect this situation to be materially changed. It was felt that the present S. A. E. Standards might be improved somewhat, in certain details, and by adding a small generator mounting, but that no action should be taken toward changing them radically. Several members of the A. E. A. Committee are also members of the Electrical Equipment Division of the S. A. E. Standards Committee, and close cooperation between the two bodies is had in this work.

In the session at which electrical servicing was discussed, it was indicated that in general the servicing of electrical equipment on automobiles should be done by authorized electrical service-stations rather than by the local car-dealers or the electrical apparatus manufacturers. This naturally led to discussion of the basis for service charges to the customer, but no one system was favored, it appearing that the "flat-rate" system has worked out well in some localities, while the "predetermined charge" has proved satisfactory in others.

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The Packard Single-Eight

By J. G. VINCENT,[1] *assisted by* W. R. GRISWOLD[2]

CLEVELAND SECTION PAPER *Illustrated with* PHOTOGRAPHS AND DRAWINGS

STATING the fundamental characteristics of the modern motor car under the headings of performance, safety, economy, comfort and taste, the authors define these terms and discuss each basic group. The specifications of the car in which the single-eight engine is installed are given, and the reasons governing the decision to use an eight-cylinder-in-line engine are enumerated.

Following a somewhat lengthy discussion of the components of engine performance, the design of the engine is given detailed consideration under its divisions of crankshaft design and the methods employed, gas distribution, the operation of the fuelizer, cylinders, valve gear and the arrangement of the accessories. Transmission design and the wearing quality of gears receive similar treatment.

Brakes and steering are subjects that are treated at some length, arguments being presented in favor of four-wheel brakes and reasons given for their adoption, together with a discussion of their effect on steering ability and of the mechanics of brake construction. Comfortable steering, axle construction, steering-gear efficiency and axle noise, are other subjects that are considered in detail. The paper is illustrated profusely and pertinent tabular data are included.

The mathematical analyses are made in appendices that will appear in the November, 1923, issue of THE JOURNAL.

UNDER the stimulus of the reaction from the great social upheaval that receded into the peace of 1918, engineers of the automotive industry began at once to adapt their product, and even the industry, to the new set of conditions prevailing. The period that followed saw many new expressions in the subsequent diversified designs, but, in all, the natural and obvious endeavor of each engineer was to interpret the new conditions and to utilize the lessons of past experience and the results of technical analysis to the best of his ability for the improvement over his previous efforts. Such is the natural order of progress. Our latest expression is in the form of the single-eight.

In the endeavor to arrive at engineering specifications for the single-eight, we tried to classify the fundamental cardinal properties of the modern motor car. These characteristics we have enumerated as being (*a*) performance, (*b*) safety, (*c*) economy, (*d*) comfort and (*e*) taste. These are not necessarily arranged in their order of importance, as that relation depends on many factors and the relative and absolute values of each determine the characteristics of the whole. I shall, however, discuss each one of these basic groups briefly and attempt to show what each comprises.

FUNDAMENTAL CHARACTERISTICS

Performance may be regarded in the restricted sense; acceleration, speed and flexibility. This is largely an element depending on the relation of the engine power to the weight of the car and the responsiveness of the engine to the throttle.

Safety refers to the inbuilt safety, strength of parts to withstand the various kinds of stress, the arrangement and functioning of the mechanical features that are instruments for control, such as the functioning of the steering-gear and the brakes.

Economy is relative, but it always connotates the minimum expenditure of either time or money for a given class of transportation. Hence, I include under this heading dependability, life, accessibility and simplicity as well as gasoline and oil economy.

Comfort denotes physical comfort, either on the part of the driver or the other passengers. This implies certain chassis and body proportions and arrangements.

Taste I define as being the personal sense for the fitness of things. It relates to appearance, finish, upholstery, quietness, smoothness and, generally, all those things that make their appeal through our sense of beauty and harmony.

The characteristics of performance, safety and economy have more or less definite standards for comparison, but the elements of comfort and taste are judged largely by personal impressions. Hence, it is more difficult to define these elements except in an arbitrary manner.

CAR SPECIFICATIONS

We gave careful consideration to each of these elements and finally arrived at the following specifications as representing our conception of the new car:

Wheelbase.—The wheelbase is to be contingent upon the space requirements for the engine when the body space from the back of the dash to the center-line of the rear axle is fixed at 88 9/16 in. for the five-passenger car and 95 9/16 in. for the seven-passenger car

Weight.—To be about 4200 lb. for the seven-passenger touring car, fully equipped and ready for the road

Tires.—The size to be 33 x 5 in. for both front and rear wheels

Acceleration.—To be from 5 to 50 m.p.h. in 18 sec. under ideal road conditions for the seven-passenger touring car with two passengers.

Maximum Speed.—To be 70 m.p.h. under ideal road conditions

Brakes.—The car to be equipped with four-wheel brakes

Engine.—To be of unit construction with clutch and transmission; inherently balanced for inertia forces and rotating couples; free from discernible forced vibrations and sensible critical speeds; simple and accessible

After deciding on these basic requirements, the next step was the selection of the type of engine.

REASONS FOR THE SINGLE-EIGHT ENGINE

Before presenting the reasons underlying the design of the single-eight engine, it must be understood clearly that one of the milestones marking the advance in en-

[1] M.S.A.E.—Vice-president of engineering, Packard Motor Car Co., Detroit.

[2] M.S.A.E.—Engineer in charge of analysis of design, Packard Motor Car Co., Detroit.

gine construction has been the progressive reduction of vibration. Vibration, or better, vibratory motion, is caused by the rapid changes or fluctuations in the magnitude or the direction of forces, these fluctuations causing motion of the particles, the component parts or even the entire engine with its foundation. These changes in the magnitude or the direction of forces may be attributable to a change in the gas pressure in the cylinder or may arise by virtue of the motion of the reciprocating and the rotating parts.

Specifically, engine vibration may be caused by an unbalance of the inertia and centrifugal forces, by the torque reaction and by the elastic distortion of the crankshaft and its supporting parts. The characteristics of these three causes are widely different.

Unbalance of the inertia and centrifugal forces is entirely an inherent condition and depends upon the number and arrangement of the cylinders and the arrangement of the throws or cranks of the crankshaft. This unbalance produces forces that are comparatively small at low speeds but that increase very rapidly with an increase of speed. At high speeds, the inertia forces are very often greater than the gas pressure on the piston. The effect of unbalance is to move the entire engine with its supports, the movement at any instant being rectilinear or rotative, the latter being caused by inertia or centrifugal couples.

The effect of the torque reaction or, rather, the fluctuation of the torque reaction, is a function of the steadiness factor or steadiness of the turning moment of the engine, which is dependent upon the frequency of the impulses and the flywheel effect of the moving parts. At slow speeds, particularly, the fluctuations are greatest; but, as the speed of the engine increases, the steadiness factor becomes increasingly higher and, at very high speeds, the fluctuations are so small as to produce practically no vibration.

Vibration due to the elastic change of shape of the bodies making up the engine is the result of using parts that do not possess enough stiffness or that are not adequately supported to make a rigid construction. Vibration of this type usually occurs in periods; that is, there are certain speeds of operation in which the impressed forces are timed with the deflection and the recoil of the parts. Such vibration is not inherent to any particular cylinder arrangement and generally can be avoided by careful design.

Of the three kinds of vibration, the unbalance that is inherent causes the greatest disturbance and is to be avoided particularly. For this reason, we specified that the only engine designs we would consider were those that are balanced inherently.

Components of Engine Performance

We may think of engine performance in the broadest sense, including not only power and activity but also economy and smoothness. An outline to show the components and relations appears as follows:

POWER

(1) *Capacity.*—Total displacement
(2) *Efficiency.*—Size of cylinders, valving and manifolding

ECONOMY

(1) *Compression-Ratio.*—Size of cylinder
(2) *Simplicity and Accessibility.* — Arrangement of cylinders, valve-gear layout, design of parts and arrangement of accessories
(3) *Gas Distribution.*—Arrangement of cylinders, firing order, manifolding and use of a fuelizer
(4) *Quality.*—Design of details, accuracy of manufacture, adaptation and quality of material, safety factors, area of wearing surfaces, lubrication and protection from foreign matter

SMOOTHNESS

(1) *Balance.*—Crankshaft and number and arrangement of cylinders
(2) *Torque Reaction.*—Number and arrangement of cylinders
(3) *Elastic Distortion.*—Number of cylinders, length of shaft, diameter of shaft, number of main bearings, crankcase rigidity, engine mounting and use of a vibration damper

We see that the cylinder arrangement enters into almost every consideration. The relative value of the cylinder arrangement, as well as the final layout of the engine, depends first upon the power requirement, because this affects the total piston displacement. Several cylinder arrangements are available to approximate the desired results, but perhaps only one is best adapted to a given size of engine. I wish to make myself clear on this point by qualifying what I mean by the best arrangement. The remainder of the chassis must, of course, be considered in order that the coordination shall give beauty of appearance and also the proper riding balance to the finished car. The best cylinder arrangement is altogether dependent on the relative values that the designer arbitrarily assigns to the major elements of power, economy and smoothness, while still another element, the cost of manufacture, would be given considerable importance for many classes of car.

Preliminary calculations showed that a total displacement of about 360 cu. in. was necessary to meet the power requirements. This, reduced to dimensions for different numbers of cylinders, gives approximately those stated in Table 1.

TABLE 1—APPROXIMATE BORE AND STROKE

Number of Cylinders	Bore, In.	Stroke, In.
6	3¾	5½
8	3⅜	5
12	3	4¼

Since an uncompromising stand toward the requirement of inherent balance was taken, it first appeared that the choice would be either the 6 or the 12-cylinder engine, the latter obviously embracing the V-type construction. On analyzing the six-cylinder design, we considered that the large size of the individual cylinder was a fundamental disadvantage, involving as it does relatively poorer efficiency, heavier reciprocating parts and heavier individual impulses, all of which are important when the speed range is to reach from 3000 to 3200 r.p.m.

In six-cylinder construction, the mass inertia of the entire engine about the point of oscillation, generally near the center of the crankshaft, does not increase at a proportional rate to the individual impulses with increasing cylinder dimensions and, therefore, the fluctuation of the torque reaction, particularly at low speeds and full power, is considerably more than with smaller cylinders. This detracts somewhat from the smooth running of large six-cylinder engines, a result that we were striving to attain. In other respects, the six-cylinder design had admirable characteristics.

Similar scrutiny of the 12-cylinder design led to somewhat different conclusions. The 12-cylinder engine is inherently balanced in inertia forces and rotating couples; and the great overlap of the power impulses

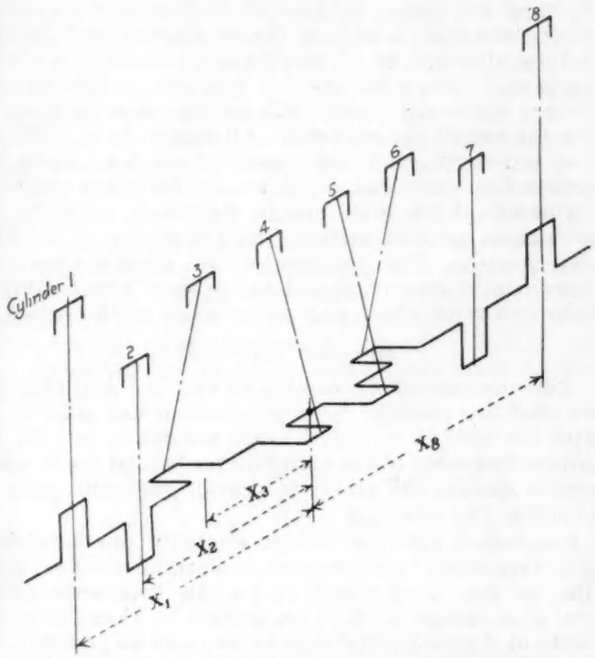

FIG. 1—GENERAL ARRANGEMENT OF THE PISTONS AND CRANKSHAFT THROWS IN THE STRAIGHT EIGHT-CYLINDER ENGINE

produces an almost constant torque, the fluctuations of the torque reaction being practically zero. The cylinder size, while comparatively small, could not be considered as a fundamental objection, yet it was felt that, after considering the average operating conditions, the fullest advantage could not be taken of such a small cylinder. The greater complication and inaccessibility of the V-type construction appeared to be the chief disadvantage. Without unduly spreading the cylinder centers, it is impossible to employ a main bearing adjacent to each crankpin, a construction that possesses great merit in the elimination of elastic distortion under the influence of the dynamic forces. This practically limits the crankshaft design to the three-bearing type, in which case the necessary rigidity can be obtained by employing a shaft of relatively large diameter for the size of the cylinder. The inaccessibility of the valves being an objection that we wished to eliminate, together with a desire for a more simple construction, led to an examination of the eight-cylinder-in-line type of engine.

An analysis of existing automobile engines of the eight-cylinder-in-line type, which were really nothing more than two four-cylinder engines with the crankshafts

[3] [This will be published in the November, 1923, issue of THE JOURNAL.]

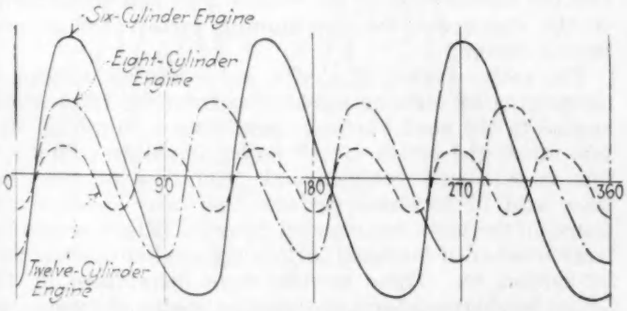

FIG. 2—TORQUE-EFFORT CURVES FOR 6, 8 AND 12-CYLINDER ENGINES OF EQUAL TOTAL PISTON DISPLACEMENT

90 deg. apart and constructed in a tandem arrangement, showed that while the inertia forces balanced so far as plane motion is concerned, a rather large couple due to the secondary inertia forces of each four-cylinder engine existed. This design was at once rejected because it did not meet the original specifications. Therefore, an examination for balance was made of other possible arrangements. It was found that the crankshaft, in which the middle four throws were arranged in a plane of 90 deg. from the outer four throws, gave a condition to the engine of practically perfect balance of plane forces and inertia couples. This crankshaft arrangement is shown in Fig. 1. Appendix 1 gives a rigorous mathematical analysis of the general case of engine balance with a special treatment for eight-cylinder-in-line engines.[3]

We consider the cylinder size to be well balanced, as our experience indicates that these dimensions occupied a rather medium position between small and large cylinders and is well adapted to the average conditions of operation, yet taking full advantage of the economical factors, such as the compression-ratio. It might appear that the fluctuation of the torque reaction was compromised, but this is not essentially true, for the frequency, as well as the magnitude of the impulse fluctuations, is an element affecting the amplitude of the vibration produced.

Fig. 2 shows the torque diagram for the three different engines; the 6, the 8 and the 12-cylinder designs that were considered for the new car. These engines are

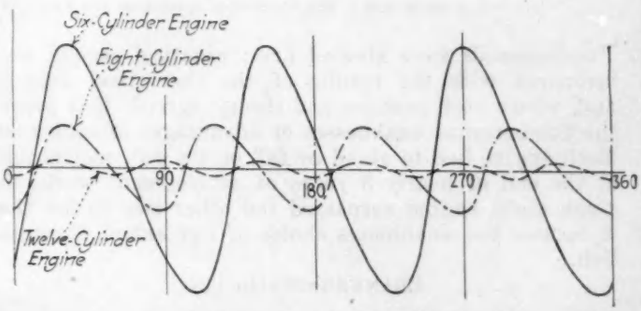

FIG. 3—RELATIVE AMPLITUDE OF VIBRATION DUE TO FLUCTUATIONS OF TORQUE REACTIONS FOR 6, 8 AND 12-CYLINDER ENGINES OF EQUAL TOTAL PISTON DISPLACEMENT

approximately of the same total piston displacement and, for comparison, the same indicator-card and the same weight of the reciprocating parts per square inch of piston-head area are used for each. This puts each engine on a comparative basis, although it should be pointed out that the six-cylinder design is favored slightly by these assumptions. The torque variations appear in the ratio of 4.0 to 2.4 to 1.0 for the 6, the 8 and the 12-cylinder designs.

The amplitude of the forced vibration varies directly as the torque variation and the square of the time during which the torque acts, and inversely as the moment of inertia of the engine and its support. The moment of inertia should be about the same for each engine. Fig. 3 is a series of curves showing the relative amplitude of these vibrations, which are in the ratio of 16.0 to 5.5 to 1.0 for the 6, the 8 and the 12-cylinder designs respectively. Thus, the amplitude for the 6-cylinder design is 16 times that for the 12-cylinder design. The fact that the 8-cylinder amplitude is 5.5 times that of the 12-cylinder design is not necessarily significant; for, while the vibration from the torque reaction from the

six cylinders is readily discernible in operation, the difference in the magnitude and the dampening influences are sufficient to nullify the vibration of the eight cylinders. This shows that no practical advantage in this respect is obtained with 12 cylinders.

The eight-cylinder design possesses the characteristics of simplicity and accessibility almost to the same degree as that of the six-cylinder, and the straight-line construction permits the use of a main crankshaft bearing adjacent to each crankpin. The length of engine was found to be very little more than that of the equivalent six-cylinder engine and provided a general chassis layout that possesses all the qualities of appearance that were desired.

To be absolutely sure of our ground and to determine whether any fallacies existed in our logic, we designed and built several cars having each type of engine. The engines were designed according to our most uptodate engineering knowledge and were built in our own shops.

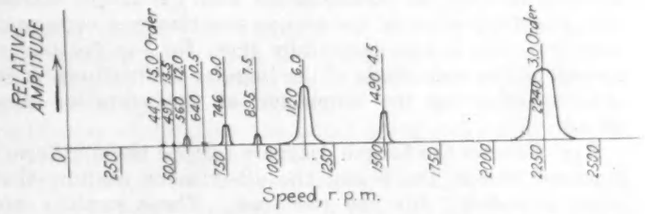

FIG. 4—RELATIVE AMPLITUDE OF TORSIONAL VIBRATIONS AT VARIOUS CRITICAL SPEEDS FOR A SIX-CYLINDER AVIATION ENGINE

Fundamentals were studied first; practical results were compared with the results of the theoretical analysis and, where both practice and theory agreed, that proved the fundamental weaknesses or advantages of each type. Each engine had to stand or fall on its own merits and, at the end of nearly 3 years of development work, the single-eight engine surpassed the other two so far that it became the unanimous choice of our entire organization.

CRANKSHAFT DESIGN

The proportioning of the crankshaft was carried out with two principal thoughts in mind, vibration and the life of the bearings. Engine vibration, as stated, depends on balance, torque reaction and elastic distortion. Balance and torque reaction having already been discussed, they can be dismissed from further consideration, and elastic distortion is the remaining element.

Vibration due to elastic distortion is most pronounced and generally only discernible when a condition of resonance occurs. The action can be described as composed of two separate parts. The impressed force due to an explosion causes, we will say, the shaft to bend out of its free position and, after the bending has proceeded to a certain point, the elastic force of the shaft overcomes the impressed force and causes the shaft to spring back or recoil in the same manner as a spring. If considerable friction happens to be present and acts as a damping influence, the shaft will resume its original shape; but, if the damping influence is small, the shaft will spring beyond the free position until recoil takes place in the opposite direction. The shaft will thus vibrate to and fro like the prongs of a tuning fork. Should the impressed forces be timed with the vibration, this is called resonance; the vibration will become stronger and stronger until the damping influence prevents any further growth.

Some damping influence is always present, either in the form of mechanical friction or friction of the particles of the material; therefore, the amplitude never becomes infinite although, in aggravated cases, the stresses in the parts may exceed the physical strength of the material. In less aggravated cases, failures may take place due to the fatigue of the material. All materials of construction are elastic, and steels, particularly hard steels, are among the most elastic. There is, therefore, some deformation of the parts due to the forces set-up by the explosions, or those arising from the motion of the various masses. The problem is to prevent sustained vibration, or resonance, and to reduce the forced vibrations to a point where they do not react on the senses.

METHODS EMPLOYED

Two methods of accomplishing this are available, but we shall see that the type of vibration has much to do with the method employed. One method is to have the natural frequency of the vibration far beyond the range of engine speeds; the other, to provide sufficient damping to nullify the vibration.

Crankshaft vibration due to elasticity can be divided into transverse and torsional vibrations. The most effective way to eliminate perceptible transverse vibration is to design a rigid crankshaft so that the movements or displacements shall be as small as possible. If the crankshaft is supported by only a few bearings in the crankcase, the size of the shaft must be very large. If many main bearings are used, the shaft can be very much smaller in diameter for a given degree of stiffness. Of course, the greatest number of bearings is obtained when one is placed between each crankpin. For an eight-cylinder-in-line engine, the number of bearings is nine. For the equivalent stiffness of a given size of shaft, with such small spans between supports as nine bearings afford, a shaft of fewer bearings would be inordinately large, and so the same degree of stiffness is not obtained in practice.

A large number of main bearings has also another distinct advantage in respect to transverse vibrations. Aside from the stiffness of the shaft, the vibration frequency bears a relation to the length of the unsupported segments of the shaft, being higher for the shorter spans; that is, all vibration which does not have nodes at the bearings is thus damped quickly, further preventing resonance. As a result of both the small amplitude and the high natural frequency of the nine-bearing crankshaft, the transverse vibration is reduced to such inconsequential effects as to be entirely beyond human sensibilities.

Torsional vibration is generally more complex than transverse vibration, because the number of main bearings has no direct influence on the mode of vibration that may take place. The torsional rigidity of the shaft, and the distribution of the masses attached or connected to the shaft, are the determining factors for a given engine design.

The entire system of shafts, gears and rotating parts comprising the driving mechanism from the front of the engine to the road surface, constitutes a vibrating system under the action of the power impulses. This system is extremely complex and, regardless of what has been said of torsional periods, they may generally be found in the most unsuspected designs. Fig. 4 shows the large number of torsional periods a six-cylinder shaft may be subject to. These periods were determined by the Geiger torsiograph for a six-cylinder engine of foreign design and illustrate the fact that periods exist even though they may be undiscernible.

For a six-cylinder engine, these periods occur when the engine speeds are one-third, one-sixth, one-ninth and the like times the primary natural vibrating frequency of the shaft; for an eight-cylinder engine one-half, one-quarter, one-sixth and the like times the frequency. The primary vibration-frequency of the shaft must always be very much higher than the maximum speed at which the engine is to operate, as synchronism here would be dangerous. The next higher order of vibration should be avoided also, for synchronism usually results in a pronounced disturbance or roughness of the engine. The next higher orders of vibration generally can be detected only by highly trained experts or by special instruments; but, even when they are not discernible, they frequently cause trouble. For instance, chronic valve-spring breakage frequently can be traced back to very slight torsional vibration.

The selection of the general dimensions of the crankshaft for the single-eight engine was fixed so that the fourth-order vibration-period occurred at a speed of about 65 m.p.h. The dimensions obtained were just sufficient to allow for the removal of the piston through the cylinder, and thereby no sacrifice in accessibility was made. At the same time, an engine design was provided that is remarkably unaffected by torsional periods. However, to provide the full protection against the insidious effects of the undetectable higher partials, the vibration damper was fitted. This device damps out the vibration in the incipient stage in a manner similar to that of a pendulum bob released to swing through a viscous fluid such as oil.

The bearing design worked in very well, as a study of Table 2 will show. Figs. 5 and 6 show graphically the loads on the various bearings at 3000 r.p.m. With pres-

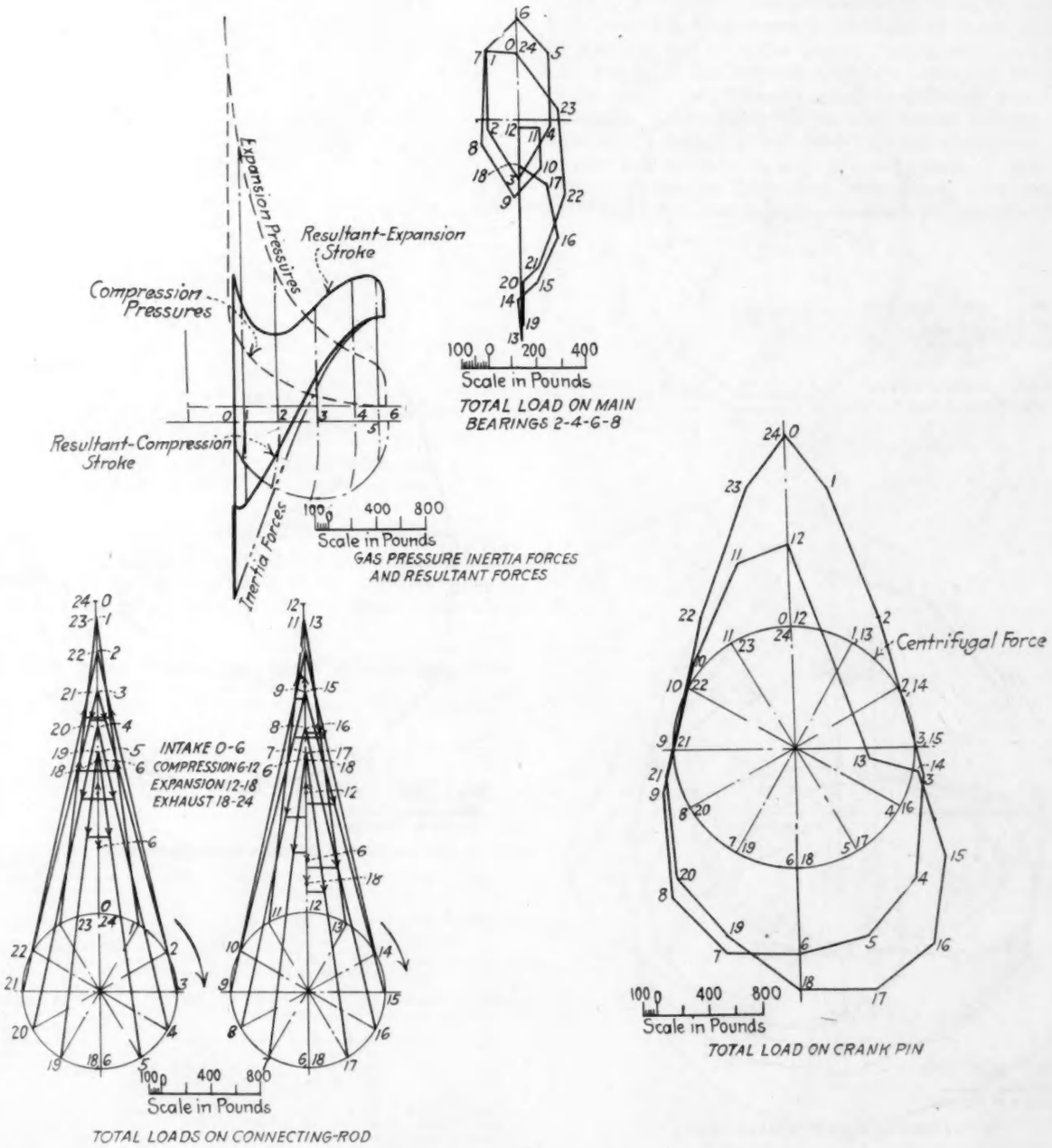

FIG. 5—DIAGRAMS OF VARIOUS PRESSURES AND LOADS

sure-feed lubrication, the unit pressure and the heating factors, *P-V* values, are fairly conservative for such a high engine-speed as 3000 r.p.m. In the proportioning of these bearings, the crankpin was favored, the *P-V* value being only 16,000 because of the metered supply of lubricant.

The intermediate center main bearings Nos. 3 and 7, and the center main bearing No. 5, operate under the most favorable conditions of any of the bearings; the oil supply is continuous and the loads are nearly constant. For these, the respective *P-V* values of 24,600 and 29,600 are within the limits of good bearing design for motor-car engines with pressure-feed lubrication.

To take advantage of every opportunity contributing to bearing life, the metering hole that feeds oil to the crankpin bearing has been given special attention. An investigation that we conducted about 2 years ago was reported under the title An Analysis of a Point in Crankpin Lubrication,[4] and led to the discovery that the most effective lubrication is obtained when the oil is introduced to the bearing on the side of minimum pressure just previous to explosion. Fig. 7 shows the pressure distribution on a crankpin during the cycle at 600 r.p.m., and also the position of the oil-feed hole in the crankpin; for 3000 r.p.m., the pressure distribution is shown by Fig. 8. As the speed increases from 600 to 3000 r.p.m., the area of greatest pressure gradually moves from the upper side around in a counterclockwise direction to the lower side of the crankpin. Normally, passenger-car engines do not operate at full power below the speed of maximum torque, which is seldom less than from 800 to 1000 r.p.m.

[4] See *Automotive Industries*, Feb. 23, 1922, p. 462.

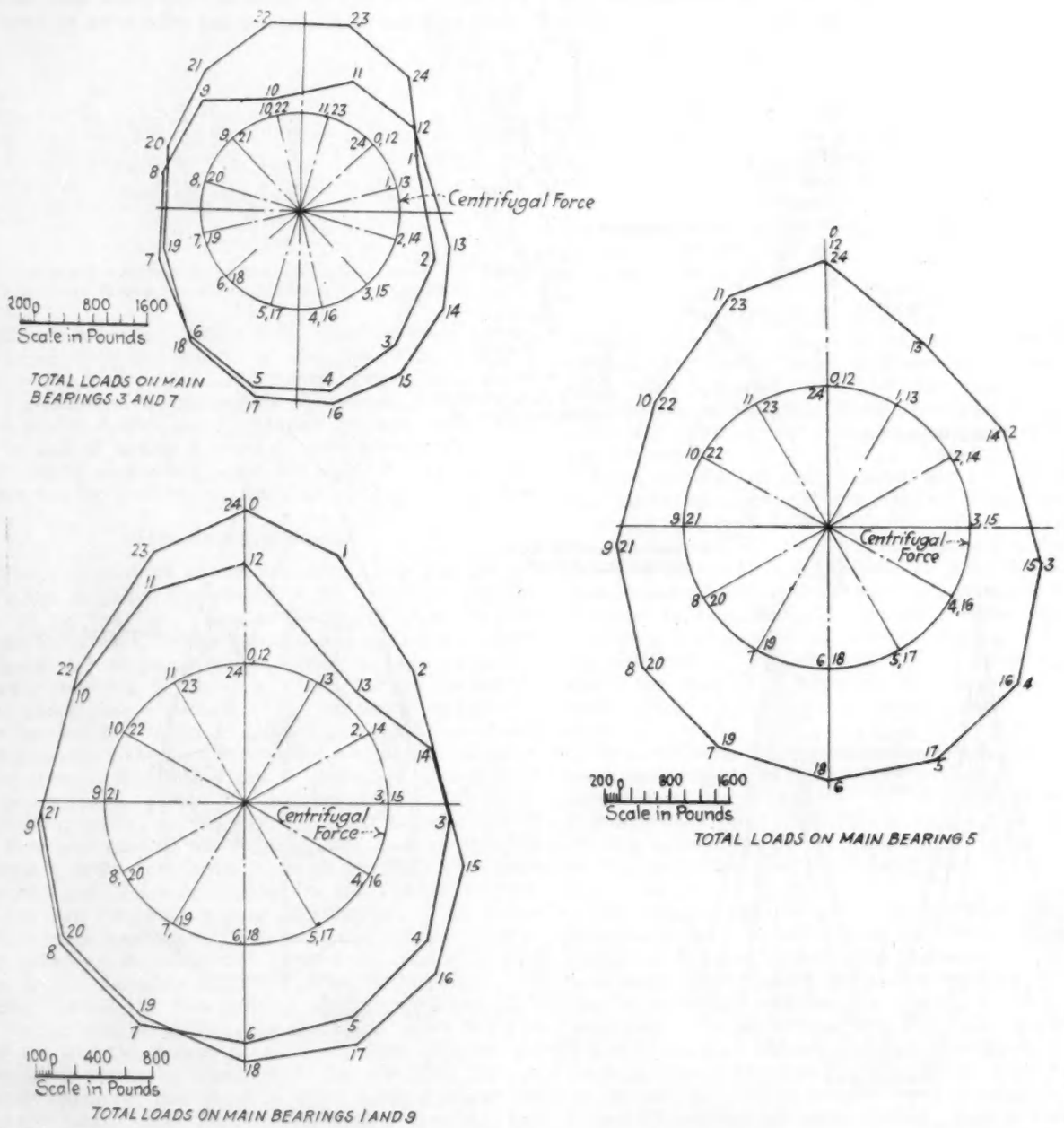
Centrifugal Force
Scale in Pounds
TOTAL LOADS ON MAIN BEARINGS 3 AND 7
Centrifugal Force
Scale in Pounds
TOTAL LOADS ON MAIN BEARING 5
Centrifugal Force
Scale in Pounds
TOTAL LOADS ON MAIN BEARINGS 1 AND 9

FIG. 6—LOAD DIAGRAMS FOR DIFFERENT BEARINGS

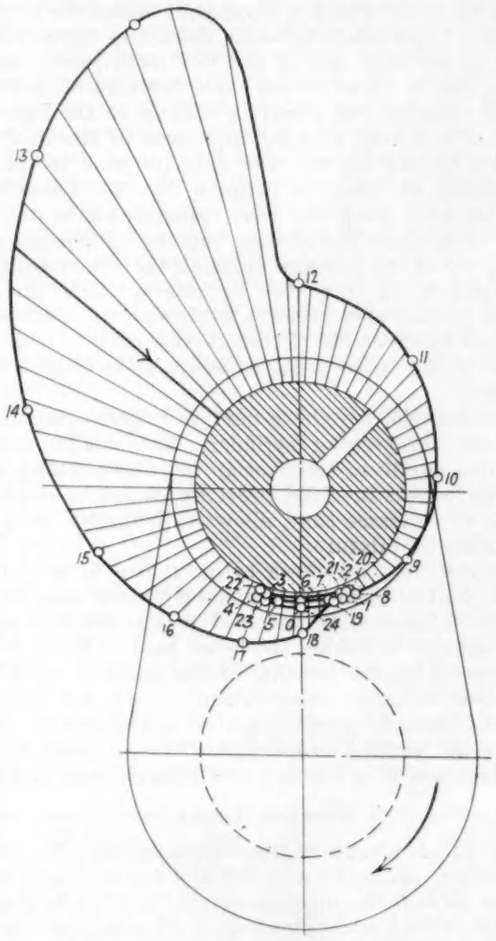

FIG. 7—DIAGRAM OF CRANKPIN PRESSURES AT A SPEED OF 600 R.P.M. AND FULL LOAD

GAS DISTRIBUTION

The original premise that eliminated the more complicated engine designs in favor of the eight-cylinder-in-line design naturally implied that a single carbureter must be adequate and must function properly. At first, the problem of distribution appeared to be a very difficult one because of the length of the passages to some of the cylinders. Uniform distribution is, at high speeds, difficult in most forms of multi-cylinder engines, and the distance to the remote cylinders of the eight-cylinder-in-line engine appeared to complicate the problem even more. However, upon a thorough study of a problem, what often appears to be a fundamental disadvantage is found not to be fundamental at all, but merely a disadvantage under special conditions. When the conditions are changed, the disadvantage often works out to be a decided advantage. This was the case with the distribution for the single-eight engine.

After an immense amount of research and experimental work, we were able to formulate a general rule that may be applicable to straight-line eight-cylinder engines, but this rule, however, appears to be fundamental. This rule is to make the gas passages as short and direct as possible, and have the engine timing arranged so that the inertia of the moving column of gas can be utilized to overcome the unequal suction in different parts of the passages, particularly when directional differences of flow exist; and, further, to utilize the maximum inertia of the gas column to aid the movement of gas to the cylinders remote from the carbureter, also supplying enough heat to the mixture to offset the radiation of the

TABLE 2—SUMMARY OF CRANKSHAFT BEARING DATA FOR MODEL NOS. 136 AND 143

	BEARINGS					
	Crank-pin	Front Main (1)	Inter-mediate Main (2, 4, (6 and 8)	Inter-mediate Center Main (3 and 7)	Center Main (5)	Rear Main (9)
Journal Lengths, Measured Overall on Shaft, in.	1½	2 11/16	1 5/16	1 11/16	1 11/16	2½
Journal Outside Diameters, in.	2⅛	2⅜	2⅜	2⅜	2⅜	2⅜
Bearing Lengths Measured Overall on Box, in.	1½	2 5/16	1 5/16	1 11/16	1 11/16	2½
Net Effective Bearing Lengths, in.	1.263	2.475	0.905	1.406	1.656	2.183
Net Effective Projected Area, sq. in.	2.685	5.810	2.150	3.340	3.930	5.180
Maximum Bearing Loads, lb. Total	2,510	2,400	880	3,205	4,400	2,400
Maximum Bearing Loads, lb. Unit	935	413	409	958	1,120	463
Mean Bearing Loads, lb. Total	1,540	1,900	357	2,645	3,750	1,900
Mean Bearing Loads, lb. Unit	474	327	166	792	955	367
P-V Values, at 3000 r.p.m. lb.-ft. per sec.	16,000	10,200	5,200	24,600	29,600	11,400

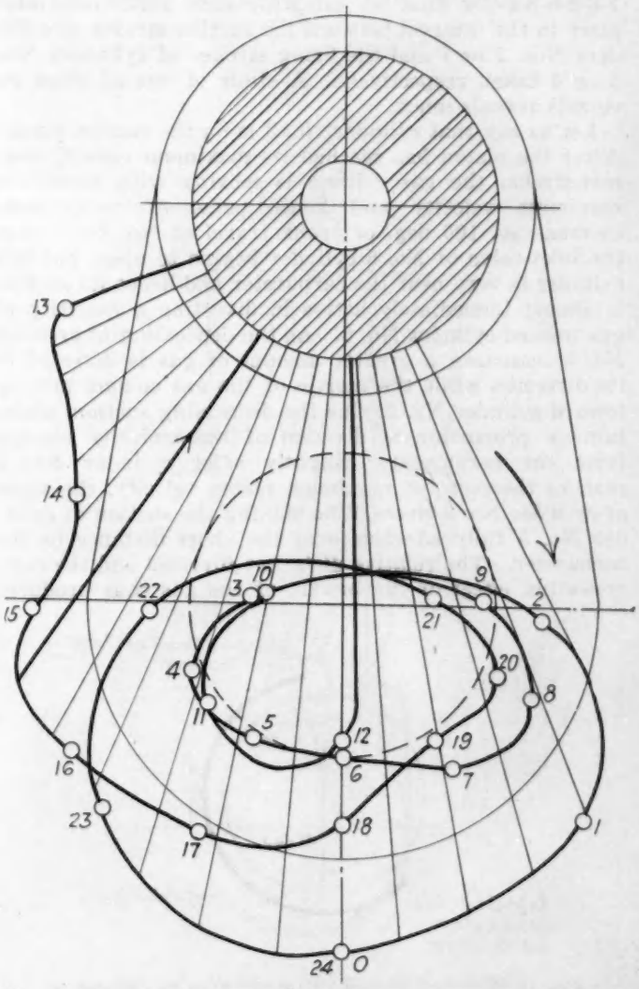

FIG. 8—CRANKPIN PRESSURE DIAGRAM AT A SPEED OF 3000 R.P.M. AND FULL LOAD

long passages. This is explained more fully by Fig. 9, which shows the arrangement of the carbureter and gas passages, and by Fig. 10, which is a representation of the inlet-valve timing for one cylinder.

For the inlet-valve opening-interval of 215 deg., it is plain that two cylinders are drawing gas in varying degrees at all times from the passage and, for very short intervals, three cylinders overlap. For convenience, let us divide the engine through the center of the passage

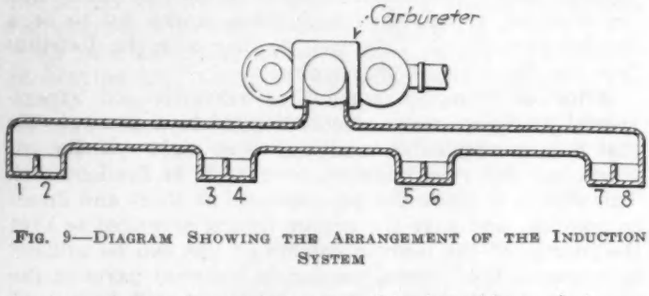

FIG. 9—DIAGRAM SHOWING THE ARRANGEMENT OF THE INDUCTION SYSTEM

leading from the carbureter; this is midway between cylinders Nos. 4 and 5. With any timing whatsoever, it is evident that the first cylinder firing in either half which follows immediately a cylinder of the other half firing must cause a change in the direction of the gas flow, from front to rear or vice versa. In the firing order 1-3-2-5-8-6-7-4, that we adopted, such a reversal takes place in the interval between the suction strokes of cylinders Nos. 2 or 7 and the firing strokes of cylinders Nos. 5 or 4 taken respectively. A study of one of these reversals reveals much.

Let us say that cylinder No. 3 is on the suction stroke. After the piston has reached its maximum velocity near mid-stroke, the gas column is moving with nearly its maximum velocity and hence possesses considerable inertia. At 100 deg. of crank travel of No. 2 cylinder, the inlet-valve of No. 5 cylinder begins to open, but this cylinder is very near the carbureter and hence its suction is almost immediately active in diverting a quantity of gas toward cylinder No. 5. As the depression of cylinder No. 5 increases, a greater amount of gas is diverted in its direction while the inertia of the gas column moving toward cylinder No. 2, plus the decreasing suction, maintains a proportionate division of the quantity coming from the carbureter. Slightly after cylinder No. 5 reaches the point of maximum piston velocity, the valve of cylinder No. 2 closes, thus making the suction of cylinder No. 5 fully effective over the short distance to the carbureter. The relative distances involved and the compensating effect of the inertia of the gas thus produces

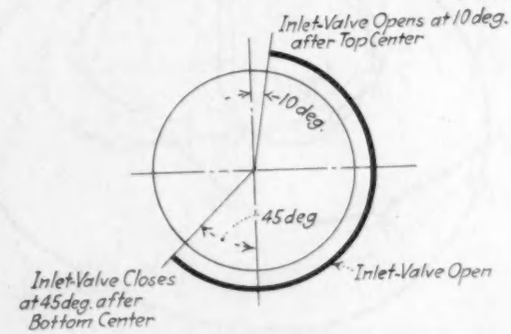
Inlet-Valve Opens at 10 deg. after Top Center
10 deg.
45 deg
Inlet-Valve Open
Inlet-Valve Closes at 45 deg. after Bottom Center

FIG. 10—TIMING DIAGRAM OF INLET-VALVE IN DEGREES OF CRANKSHAFT TRAVEL

a condition in which a practically equal distribution of mixture takes place between these two cylinders.

Next, we have one of the two most remote cylinders firing, No. 8 cylinder. At the conclusion of induction to No. 5 cylinder, the complete change in the direction of the flow has been effected, and some of the mixture has already passed beyond this cylinder, due to its inertia in making the bend to cylinder No. 5. Therefore, the only work of induction that cylinder No. 8 has to perform is to draw the already moving column of mixture to its end of the passage, through the intervening length of manifold of only two cylinders. This intervening length is desirable because, otherwise, if cylinder No. 8 followed cylinder No. 7, the inertia of this long column of gas would be so great as to cause loading of the end cylinder.

Cylinder No. 6 follows next in order. Here we have only one intervening cylinder. The double compensation, due partly to the overlap in valve opening and the passage lengths involved, prevents loading of cylinder No. 8 and, at the same time, allows cylinder No. 6 to get its full share of mixture.

Cylinder No. 7 follows cylinder No. 6, and it would appear that cylinder No. 7 would get more than its share; but the compensation involved in the reversal of direction that occurs during the latter part of the valve opening, caused by the opening of the valve of cylinder No. 4, causes an equal distribution again. Of course, the difficulty caused by condensation might prove serious if it were not handled properly, but this is overcome by the combined use of a fuelizer and exhaust-manifold heat.

FUELIZER OPERATION

Fig. 11 shows the fuelizer installation. In operation the fuelizer heats the gasoline and the air from the carbureter so that the mixture enters the cylinders as a dry gas rather than a combination of air and liquid particles. This is accomplished by shunting a small fraction of the gasoline and air mixture passing to the engine through the throttle-valve shaft into a passage in the manifold that leads to the burning chamber of the fuelizer. This mixture is ignited by the spark-plug in the side of the burning chamber of the fuelizer and burns with a steady flame, which can be viewed through the inspection glass on top of the fuelizer. The flame heats the burning-chamber walls and then the hot burned gas mixed with the charge going to the engine from the carbureter. The application of heat to the ingoing mixture improves its gasification greatly, which is particularly important at the lower throttle-openings, at which time the fuelizer supplies its maximum heat. With an increasing engine speed, the fuelizer heat gradually diminishes and, at high speeds where this heat is no longer required, the fuelizer automatically ceases to function.

The total quantity of gasoline for a given setting of the carbureter is little affected one way or the other by the gasoline burned in the fuelizer, but the fuelizer makes possible the use of a mixture that greatly increases the economy. In this way when the manifold is designed properly, when the firing order is arranged to take advantage of the inertia effects of the gases and when sufficient heat is supplied to maintain a dry mixture, the uniformity of distribution is as nearly effected ideally as it is practically possible to make it.

CYLINDER AND VALVE-GEAR

Of necessity the cylinder and valve-gear layouts are complemental. The engine thermal and volumetric efficiencies are dependent very largely on the design of the

clearance or combustion space of the cylinder. The design of the combustion space is influenced greatly by the location of the valves referred to the cylinder proper. Fundamentally, the radiating surface of the combustion-chamber should be as small as possible for the given volume of the clearance required. Also, the gas flow into the cylinder should be direct. For the L-head cylinder, this is accomplished in part by placing the valves close to the cylinder bore. With direct-acting tappets, the camshaft must be placed in line with the valve; this brings the camshaft close to the center of the engine and, to provide clearance for the lower ends of the connecting-rods, the crankshaft must be placed far enough below the camshaft. This requires long connecting-rods and, therefore, makes for a high and consequently, for a heavier engine.

Using rocker levers allows the axis of the camshaft to be placed farther from the center of the engine and, therefore, permits a more compact and lighter engine construction. For a given diameter of camshaft, better shaped cams are obtained for a given valve-lift and acceleration; that is, the static component, equivalent to the side-thrust, of the cam-force acting on the rocker-arm, is of a small proportion to the total lifting force. The reason is that the base-circle diameter is larger in proportion to the lift of the cam. This static component is reacted at the rocker-pivot, which is an oscillating pin bearing and generally is considered superior to bearings sliding along straight elements. This can be explained by analyzing the relation of forces acting on the rocker-lever and the ordinary follower. In Fig. 12, the cam is shown turning clockwise and lifting the rocker-lever against a force F_t at the tappet. The lifting force required is, say, F_l, acting in the direction shown.

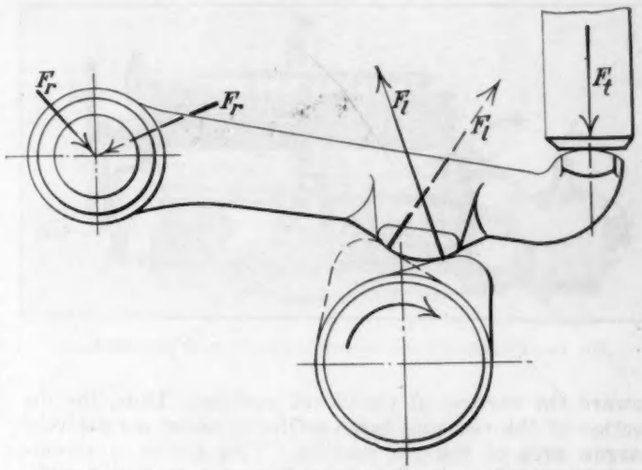

FIG. 12—DIAGRAM OF THE FORCES ACTING ON THE ROCKER ARM

The reaction at the pivot is shown by the force F_r. As the cams revolve farther, the direction of the force F_l approaches the vertical, with a consequent approach to the vertical of the reacting force F_r. At the maximum lift, these forces are vertical and, with further cam rotation, the forces incline in the other direction again

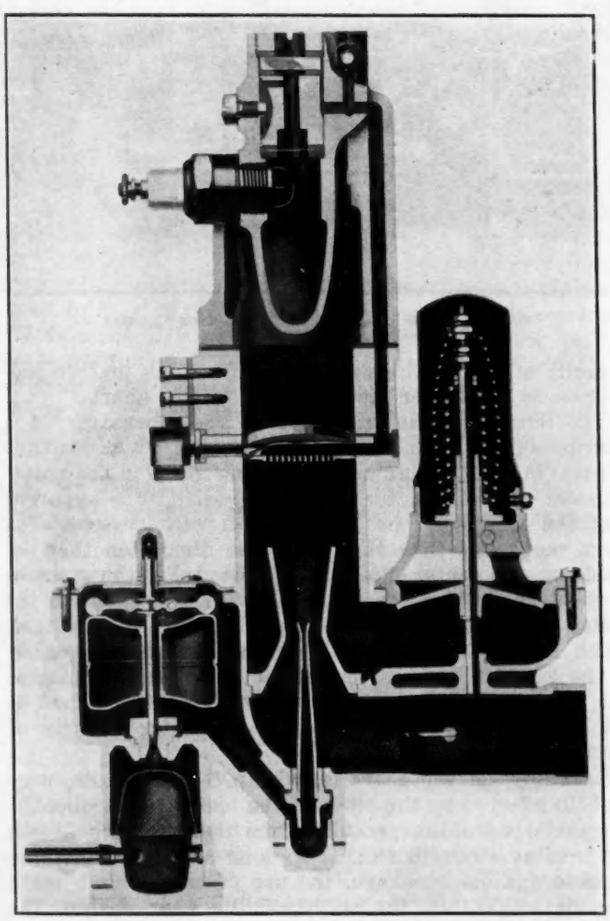

FIG. 11—SECTIONAL VIEW OF THE CARBURETER AND FUELIZER

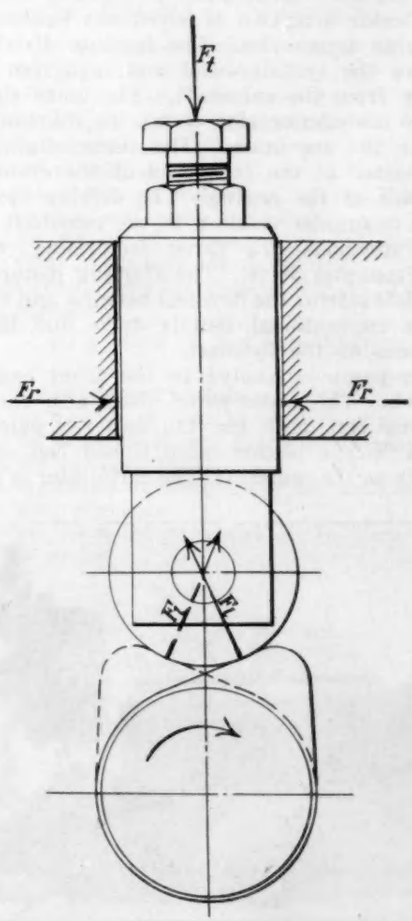

FIG. 13—DIAGRAM SHOWING THE FORCES ACTING ON AN ORDINARY ROLLER TAPPET

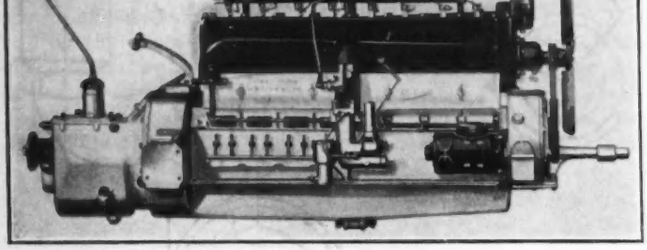

FIG. 14—EXTERIOR VIEW OF THE RIGHT SIDE OF THE ENGINE

toward the vertical at the closed position. Thus, the direction of the reacting force oscillates about a relatively narrow area of the pin bearing. This action is accompanied by no diametral change of the clearance and without impact. With the ordinary roller tappet shown in Fig. 13, there is a take-up of the clearance across the diameter with every change of the direction of the side-thrust. At high speeds, with worn guides, the impacts produced cause considerable noise. Thus, with the cylinder and valve-gear construction that has been adopted, we feel that ordonnance, a right arrangement of parts to produce the best effect, has been accomplished.

ARRANGEMENT OF ACCESSORIES

To carry out the original plan of simplicity and accessibility, particular attention is given the location of the various engine accessories. The ignition distributor is placed above the cylinder-head and is driven through spiral gears from the camshaft. The same shaft that operates the distributor also drives the oil-pump, which is located in the crankcase. The electric-lighting generator is located at the front end of the crankcase on the right side of the engine. The driving sprocket is placed in a triangular relation to the camshaft and the crankshaft sprockets, the three comprising, with the chain, the front-end drive. The starting motor is situated at the left side of the flywheel housing and functions through the conventional Bendix drive and the alloy-steel ring-gear on the flywheel.

The water-pump is located in the front end of the cylinder-block. The water-pump shaft also carries the air-circulating fan, both the fan and the pump being driven by a V-type leather-faced linked belt operating from a pulley on the camshaft. The carbureter is mounted on the right side of the engine, at the middle of the length of the cylinder. Fig. 14 shows the right side of the powerplant and Fig. 15 shows the cooling system viewed from the left side of the engine. The simplicity of the entire arrangement is evident.

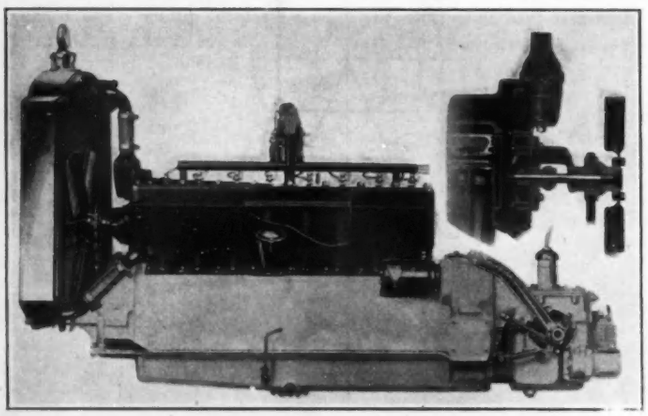

FIG. 15—EXTERIOR VIEW OF THE LEFT SIDE OF THE ENGINE SHOWING THE COOLING SYSTEM

TRANSMISSION DESIGN

The construction of the transmission and that of the clutch are shown in Fig. 16. Economy of transmission design depends very largely upon the proper coordination of the gear design and upon the sizes of the shafts and the bearings. This is dependent largely upon the center distance of the shafts being in a proper relation to the capacity of the transmission. The capacity of a gear varies approximately as the square of the diameter and directly as the effective face-width for a given pitch and pressure angle. On the other hand, for a given transmission capacity, the shaft size varies in a direct proportion of the length between the supports or bearings and inversely as the fourth root of the center distance. The capacity of the bearings must increase in an inverse proportion to the center distance between the shafts.

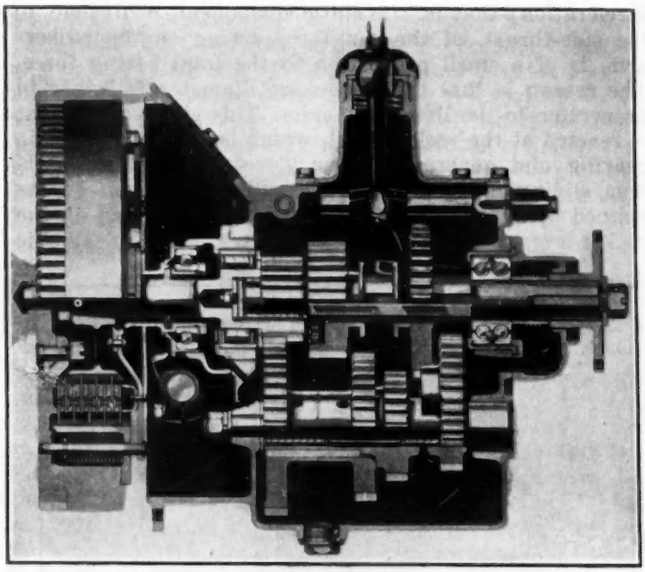

FIG. 16—THE TRANSMISSION AND THE CLUTCH

The length of the transmission case increases in a multiple of the increase in the face-width, which multiple is equal to twice the number of sliding gears in the transmission. Therefore, for a given capacity, it is apparent that the length of the transmission will increase at a more rapid rate than the transverse dimension that depends on the center distance. Incidentally, the stresses in the transmission case increase with a decrease in the center distance; therefore, the stiffness of the case, which has an appreciable effect on the running quality of the gears, also is associated with the center distance. The most economical coordination cannot be obtained as yet by any formulated rules but only after a study of specific conditions.

Durability of the gears is, within certain limits, very slightly affected by the pitch of the teeth. Consequently, no special advantage results from designing the teeth for bending strength alone. As long as ample provision is made against breakage, the use of coarse-pitch teeth is undesirable. In the study of the gear design, the pitch, the number of teeth in simultaneous contact,

the specific sliding factors and the pressure angle are elements requiring consideration. In general, tooth failures rarely occur in present-day practice. The compressive strength between two teeth in contact under load seems most important; that is, the surface pressure of the opposing curved tooth-faces must not exceed the endurance limit of the material because, otherwise, surface fatigue will take place eventually and cause changes in the tooth-form that greatly impair the running quality or quietness of the gears.

The compressive strength also has a direct influence on the preservation of the lubricating film between the teeth because, the greater the surface pressure is, and the smaller the relative velocity of sliding of the teeth is, the more difficult it is to maintain the oil-film. The relative sliding speed is zero at the pitch-line and, except for the sliding produced by the minute deformation of the surfaces in the contact, pure rolling of the teeth occurs. At this point the oil is squeezed out easily if certain unit pressures are exceeded.

Wearing Qualities of Gears

We can compare the wearing qualities of gears made of the same material identically heat-treated by the modified Hertz formula

$$K = P\,[(1/r_1) \pm (1/r_2)] \div b \sin 2\phi \qquad (1)$$

where

ϕ = the pressure angle
b = the face width
K = a factor proportional to the maximum compressive stress
P = the total tangential load on the teeth
r_1 = the pitch radius of the pinion
r_2 = the pitch radius of the gear

This formula applies strictly only within the compressive elastic-limit of the material, but it is comparative for stresses exceeding this limit. It has been found that, for a limited number of applications of load, the value of K may greatly exceed the permissible value for continuous operation of equal durability. For this reason, the values of K may vary over a considerable range for the different speeds, being lowest for the constant-mesh gears and greatest for the reverse gears. The tooth load P, computed from the maximum engine-torque, is used in making comparisons and the permissible value of K depends very much on the materials and the heat-treatments for steels, larger values generally accompanying greater hardness.

Wear of the gear teeth is also a function of the relative sliding that takes place between the surfaces of contact, while quiet operation depends to an appreciable extent on the number of pairs of teeth in simultaneous contact; or, upon the overlap between successive engagements. These two elements conflict in tooth-forms that have the addendum equal to the dedendum.

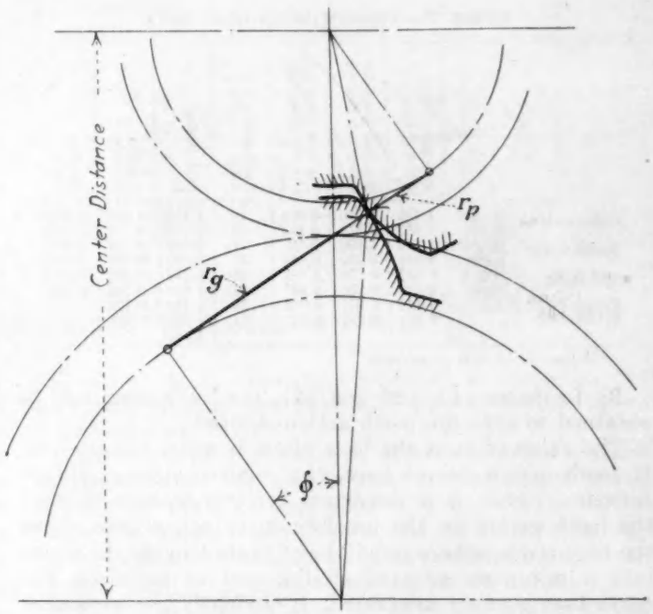

Fig. 17—Diagram for Determining the Radius of Curvature of the Involutes at the Point of Contact of the Gear Teeth

The relative sliding is also a function of the pressure angle and may be expressed by the formula

$$\left.\begin{aligned} Q_p &= 1 - r_g/r_p R \\ Q_g &= 1 - r_p R/r_g \end{aligned}\right\} \qquad (2)$$

where

Q_p = the relative sliding for the pinion
Q_g = the relative sliding for the gear
R = the velocity ratio
r_g = the radius of curvature at the instantaneous point of contact for the gear
r_p = the radius of curvature at the instantaneous point of contact for the pinion

Reference should now be made to Fig. 17. The number of teeth in contact is

$$N_c = \tan\phi\ (n_1 + n_2)\ (r_b - r_e)/2\pi \qquad (3)$$

where

N_c = the number of teeth in contact
n_1 = the number of teeth in the driving gear
n_2 = the number of teeth in the driven gear
ϕ = the pressure angle
r_b = the radius of curvature of the tooth at the beginning of action
r_e = the radius of curvature of the tooth at the end of action

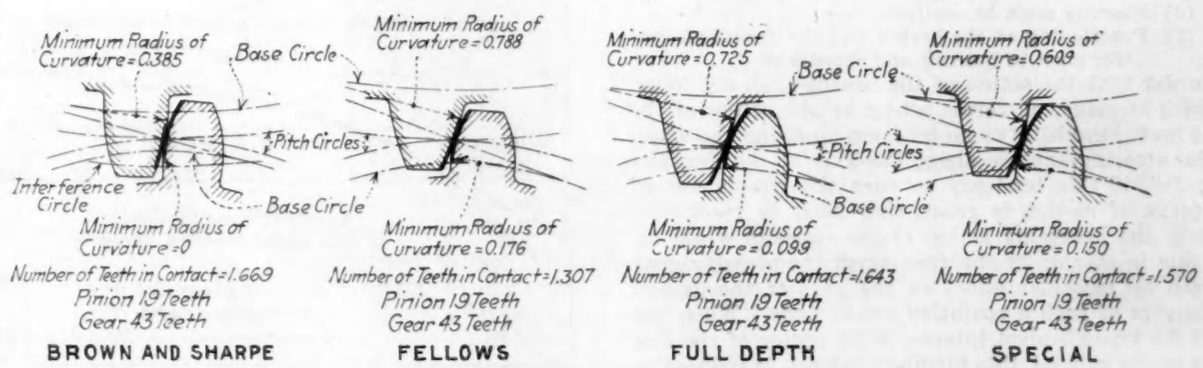

Fig. 18—A Comparison of Various Tooth Profiles

TABLE 3[a]—TRANSMISSION-GEAR DATA

		Maximum Overall Width of Face, In.	Net Face Width for Computing Flexture Stress, In.	Net Face Width for Computing Bearing Stress, In.	Number of Teeth	Pitch Radius, In.	Strength Factor	Circular Pitch
Constant-Mesh	Pinion	1.500	1.000	0.812	19	1.357	0.098	0.4488
	Gear	0.812	0.812	0.812	43	3.070	0.113	0.4488
Second-Speed	Pinion	1.375	1.000	0.687	27	1.930	0.110	0.4488
	Gear	0.812	0.750	0.687	35	2.500	0.103	0.4488
First-Speed	Pinion	0.812	0.750	0.687	25	1.790	0.104	0.4488
	Gear	0.812	0.750	0.687	37	2.640	0.106	0.4488
Reverse-Speed Pinion		0.812	0.812	0.763	20	1.440	0.104	.....
Reverse Idler		0.812	0.781	0.687	24	1.730 1.710	0.106	.....

[a]All gears have 0.4248 normal pitch

TABLE 4—CALCULATED STRESSES

		Mean Tangential Tooth Load, Lb.	Mean Tooth Load per Inch of Face Width, for Bending, Lb.	Tooth Stress Flexture Lb per Sq. In.	Mean Tooth Load per Inch of Face Width, for Bearing, Lb.	Relative Curvature	Bearing Stress Factor K
Constant-Mesh	Pinion	2,010	2,010	30,500	2,480	0.606	4,300
	Gear	2,010	2,480	34,400	2,480		
Second-Speed	Pinion	2,470	2,470	68,300	3,590	0.702	5,380
	Gear	2,470	3,300	55,100	3,590		
First-Speed	Pinion	3,460	4,610	94,800	5,030	0.686	7,850
	Gear	3,460	4,610	96,800	5,030		
Reverse-Speed Pinion		4,330	5,320	114,000	5,550	0.550	.....
Reverse Idler		4,330	5,530	115,000	5,550	.....	13,100
Reverse-Speed Gear		4,330	5,770	122,000	6,300	0.670	.....
Reverse Idler		4,330	5,530	115,000	6,300	.....	10,100

By formulas (1), (2) and (3), the tooth-form can be obtained to give the tooth action desired.

The value of r_b at the base circle is equal to zero and, if tooth action begins here, the relative sliding will be infinite. Hence, it is important for two reasons to start the tooth action on the involute curve somewhere above the base circle, where r_b will be of finite length: to maintain a minimum rate of sliding and to maintain the curve that is easily generated. It obviously is impracticable to generate with accuracy a curve that has a radius of curvature equal to zero. Figs. 18 and 19 show the improvement that can be effected by varying the specific sliding, the number of teeth in contact and the durability factors. Tables 3 and 4 give the particulars relating to the single-eight transmission-gear design.

Accuracy and finish are also elements contributing to durability and running quality. Generally, the more accurate the profiles are and the smoother the finish at the beginning of service is, the greater the durability against wear will be. So far, finish-grinding the tooth profiles offers the most accurate method of reproduction. We have adopted this for regular production.

Fundamental Requirements of Brakes and Steering

Faster driving speeds and greater accelerations require more rapid and more effective speed reduction. In large cars of high power, the maximum degree of retardation for two-wheel brakes has been reached and, therefore, to improve braking beyond this point, four-wheel brakes must be employed. Good four-wheel brakes require careful study in their design and must be carefully made, for hazardous results follow if either the design or the workmanship is subordinated to cost or if they possess defects due to insufficient study. The fundamental requirements are summed up as follows:

(1) Action of the brakes must not interfere with steering
(2) Brakes must be equalized properly at all times
(3) Brakes must operate with ease and directness
(4) Steering must be comfortable
(5) Functioning of the brakes and the front-axle system must be positive and dependable

In order that the action of the brakes shall not interfere with steering, all forces set-up by the action of the brakes must have little or no tendency to swing the wheel and the steering-knuckle about the axis of the knuckle-pivot; for, if this tendency to turn the wheels out of their plane of motion is great, they must be restrained either by the balancing action of the opposite wheel or, when this is absent, by the friction in the steering-gear and even by muscular effort on the part of the driver. The dangers of such a condition are obvious. When the axis of the knuckle-pivot intersects the center of the tire contact at the ground, this turning tendency is reduced to the minimum.

To explain the action that braking forces have on steering, the following analysis is given. A correct analysis cannot be made based on the assumption that the steering-spindle and the wheel can be treated as a rigid body. The wheel and the knuckle must be treated separately; the forces and their reactions must be determined for each. Another element that has been the cause of considerable confusion is the failure to distinguish between the different kinds of resistance that can occur between the tire and the road. This is absolutely essential to a correct analysis of the problem.

For a car with a high degree of freedom for the rotation of the road wheels, one having anti-friction bearings and moving in a straight path on a horizontal plane which, in a practical sense, means moving straight ahead

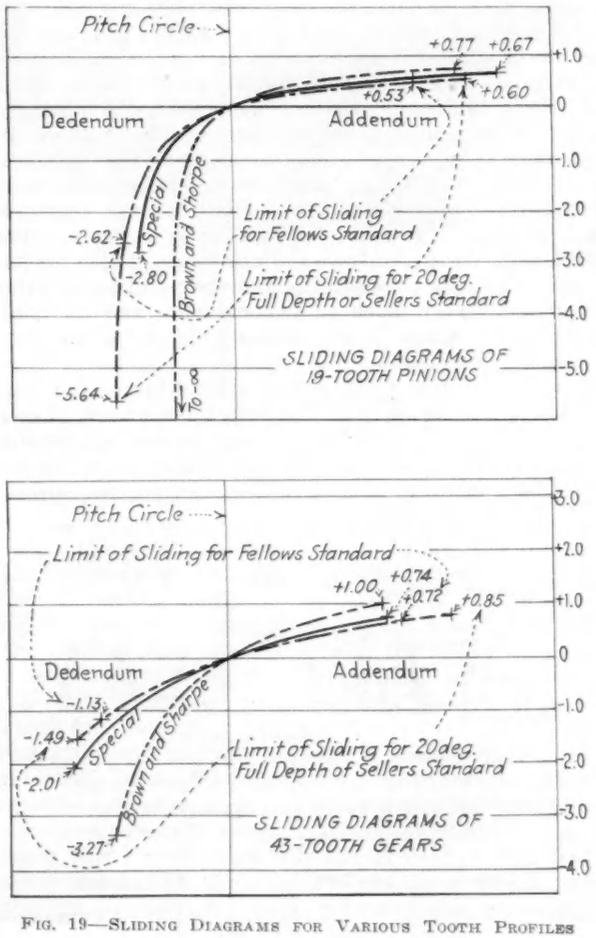

Fig. 19—Sliding Diagrams for Various Tooth Profiles

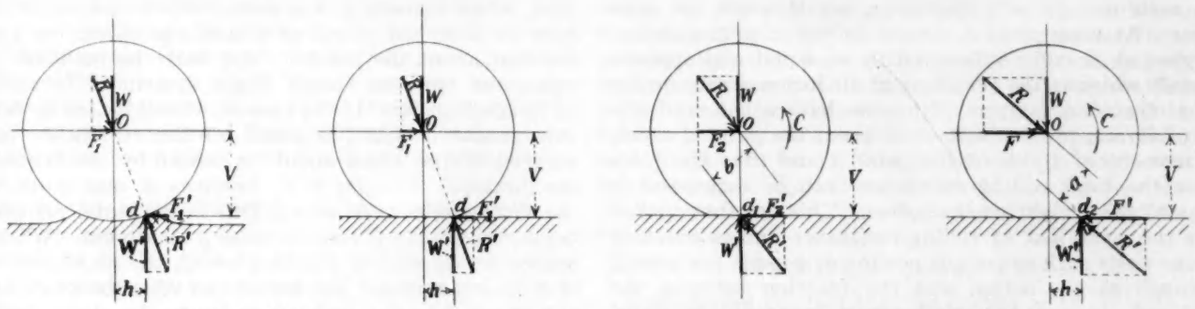

FIG. 20—DIAGRAMS OF THE VARIOUS FORCES INVOLVED IN THE ROTATION OF A WHEEL

on a level road free from deviating forces such as may arise from ruts, the resistance to the motion of the wheels by the road is due to the rolling-resistance and has a definite value for any combination of tire and road conditions. With the car again in rectilinear motion as before, but with the freedom for the rotation of the wheel sufficiently reduced as by locking the wheels with the brakes, we have the condition of the tire sliding on the road. The resistance to motion is then due to sliding friction which, for the same conditions of the tire and the road surface in contact, is many times greater than the rolling-resistance, in some cases being more than 100 times greater.

A third condition may occur in which the resistance to motion may be a maximum. Assume, as before, the car in rectilinear motion and the front-wheel brakes gradually applied until they become locked. At any point in this motion, while the wheels are turning, the resistance to their motion by the road is due to a combination of rolling-resistance and static friction, the amount of resistance due to static friction depending on the resistance to the turning of the wheels by the brakes, becoming a maximum on hard road surfaces just at the instant the wheels begin to slide on the road. On certain soft road-surfaces, the resistance may be a maximum after the wheels are locked. In general, the coefficient of static friction is greater than that of sliding friction but, as actual quantitative values are not essential to this discussion, we may even assume them to be the same. It is absolutely essential to recognize that, qualitatively, they are alike and different from the rolling-resistance.

That the total resistance to motion becomes markedly decreased when the wheels become locked is a common experience to motorists, especially on wet pavements. When the wheels begin to slide, the car seems to go right on, the rate of slowing down being reduced greatly. This is due, partly, to the loss of the rolling-resistance and partly to the lower value of the kinetic or sliding friction.

WHEEL FORCES AND ROLLING-RESISTANCE

In regard to the forces that act on a wheel to overcome rolling-resistance, let us consider a wheel, as in the left half of Fig. 20, in uniform motion on a horizontal plane and overcoming rolling-resistance. The theory of rolling-resistance is based on the assumption that the surfaces in rolling contact undergo a certain amount of compression at the place of contact; hence, the effect is the same as if the roller were continually being moved out of a hollow or up on the edge of a flat on the periphery. If the road is soft compared to the roller, the effect will be that of moving out of the hollow; if the rim of the roller is very soft, as is true of an automobile tire when compared to the road, the effect is that of moving up on the edge of a flat.

A certain force, say F_1, applied at the center O of the wheel is necessary to roll the wheel at a uniform speed. The weight W also acts through the center of the wheel.

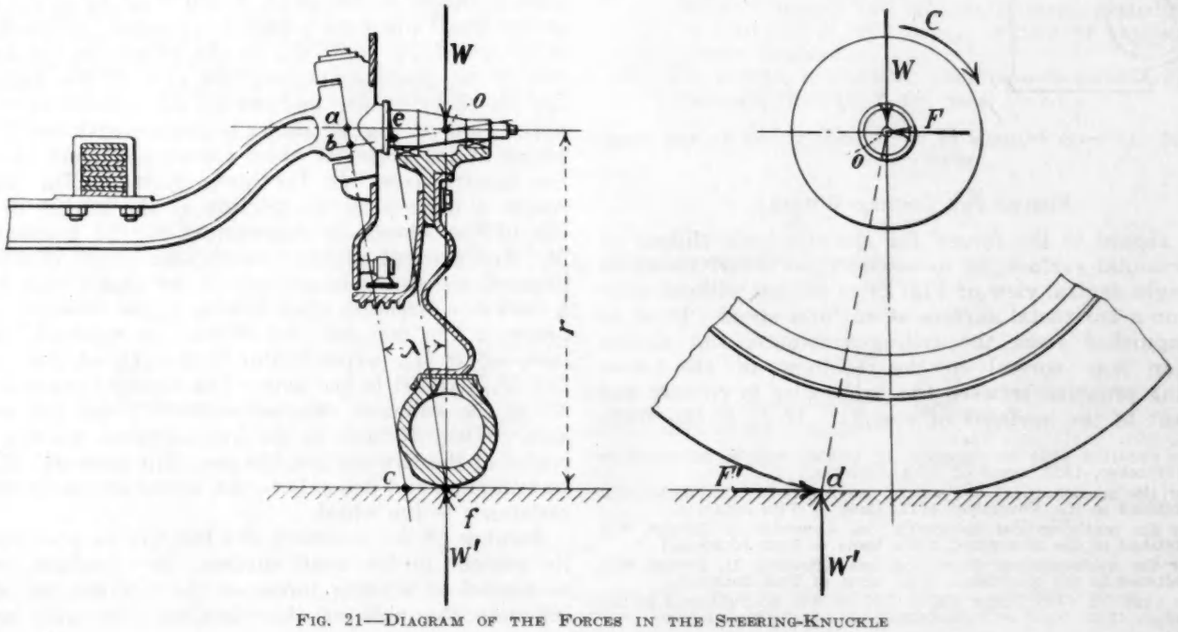

FIG. 21—DIAGRAM OF THE FORCES IN THE STEERING-KNUCKLE

The resultant, R, will, therefore, act through the same center. At some point d, where its line of action strikes the ground, it will be balanced by an equal and opposite force R' which is the resultant of all forces acting on the tire at the road surface. Since we have a balanced system of forces, the moment of W about the point d equals the moment of F_1 about the point d and thus the force F_1, or the total rolling-resistance,[5] can be expressed in terms of the weight on the wheel. This is often spoken of as the coefficient of rolling-resistance and is given in various units such as pounds per ton or pounds per pound. Inasmuch as the action and the reaction between the wheel and the steering-spindle must be equal, the wheel can be removed and replaced by the forces W and F_1 acting through the same point on the spindle. This will be done throughout the following analysis.

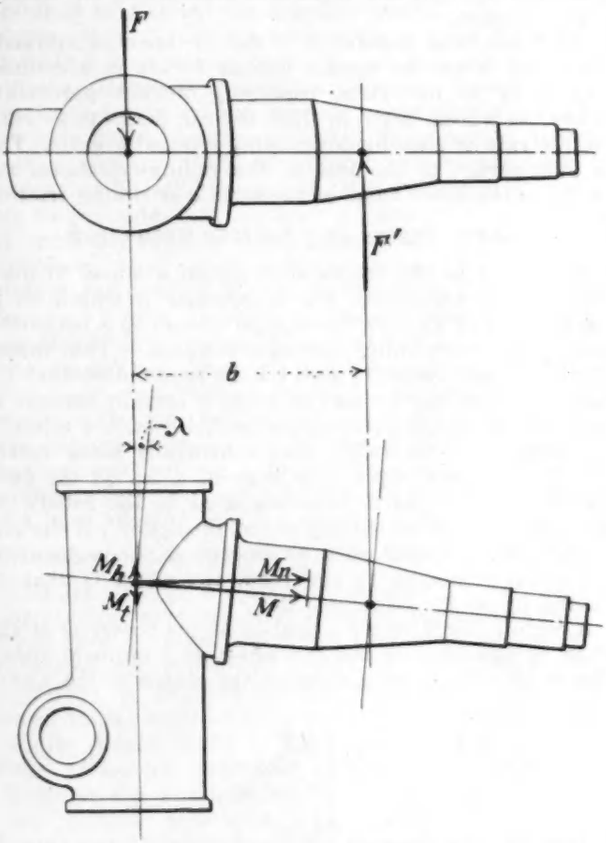

FIG. 22—ANOTHER DIAGRAM OF THE FORCES ACTING ON THE STEERING-KNUCKLE

FORCES FOR LOCKED WHEELS

In regard to the forces[6] for locked wheels sliding on a horizontal surface, let us consider the wheel shown in the right central view of Fig. 20 as sliding without rotation on a horizontal surface at uniform speed. Here, as distinguished from the rolling-resistance, the sliding friction acts normal to the resultant of the forces causing pressure between the two bodies in contact and tangent to the surfaces of contact. If F_2 is the force that, when applied to the center of the wheel, is necessary to slide the wheel at a uniform speed, we have a moment about the point d_1 that must be resisted by an equal and opposite couple if the system of forces is to be in equilibrium. In the case of wheels locked by brakes, this couple C must be equal to the resistance to the turning of the wheel about its center by the friction of the brakes. The force F_2 becomes a maximum when the sliding velocity is zero. This is the static resistance; hence, the couple C also becomes a maximum. When the brakes are applied to a rolling wheel, the maximum value of F_2 is a maximum just before the wheel becomes locked and is equal to the resistance due to the static friction. If we remove the wheel from the spindle and replace the forces W and F_2 and the couple C acting on the spindle, the system of forces will not be disturbed.

ANALYSIS FOR MIXED RESISTANCE[7]

When the wheel is rolling on a horizontal plane and a tendency is exerted on it in such a way as to offer resistance to rotation, the resistance to uniform motion is a mixed resistance due partly to the rolling-resistance and partly to the static friction. Experiments show that, under this condition, the horizontal force which, when applied at the center of the wheel, is necessary to maintain uniform motion, may be so great as to equal the total weight on the wheel.

Let us consider the wheel shown in the extreme right view of Fig. 20. The horizontal force F is equal to the sum of the rolling-resistance and the static resistance to the wheel. The reactions at the tire will act at the point d_2 off of the vertical center-line so that the couple C, which must be applied by the friction of the brakes to resist the rotation of the wheel, is equal to the moment of static resistance F_2 about the point d_1. Here again, the wheel may be removed and the forces acting at O be replaced by equal forces and couples, and the resistance will not be disturbed.

MOMENT ABOUT THE KNUCKLE-AXIS[8]

Fig. 21 shows diagrammatically in the left drawing the arrangement of the front-axle construction in which the center-line of the steering-knuckle pivot makes any angle, λ, with the center plane of the wheel. The right portion shows the forces W, F and C acting at the center of the wheel when the brakes are applied. Since the line of action of the force due to the weight on the knuckle lies in the plane containing the axis of the spindle, it has no effect on the moment of the spindle axis. The upper view of Fig. 22 shows a knuckle with the force F acting at the center O which causes a moment M_h about the knuckle-axis. In the lower view of Fig. 22, the couple C offered by the friction of the brakes to turning of the wheels, is represented by the moment axis M. The moment axis is a vector, the length of which is proportional to the magnitude of the couple that acts in a clockwise direction when looking in the direction of the arrow. The moment axis M can be resolved into its components M_n, perpendicular to the axis of the knuckle, and M_t, parallel to the axis. The algebraic sum M_h and M_t is the value of the net moment about the spindle axis. When the axis of the knuckle-pivot intersects the center of the tire contact, the resultant moment[9], M_r, will be minimum and due only to the forces set up by rolling-resistance to the wheel.

Because of the necessity for the tire to accommodate its contact to the road surface, the resultant of the tangential or braking forces on the tire will not always intersect the axis of the steering-pivot with center-

[5] See equation (35) in Appendix II, [which will be published in the November, 1923, issue of THE JOURNAL.]

[6] For the mathematical discussion, see Appendix II, [which will be published in the November, 1923, issue of THE JOURNAL.]

[7] For the mathematical discussion, see Appendix II, [which will be published in the November, 1923, issue of THE JOURNAL.]

[8] For the mathematical discussion, see Appendix II, [which will be published in the November, 1923, issue of THE JOURNAL.]

[9] See equation (40), Appendix II, [which will be published in the November, 1923, issue of THE JOURNAL.]

point design; but, under certain conditions, it will act with a small moment-arm about it. However, the theoretical arrangement represents the normal condition and the median from which slight departures may occur in any direction. Experience has shown further that, with this type of steering-layout, the action of the brakes even under abnormal conditions does not interfere with steering.

If, now, the brake-actuating mechanism has a fixed center of motion for the entire front-axle and brake construction, the brakes will be unaffected by the steering when the wheels are turned out of the straight ahead position.

Brake Equalization

Requirement (2), embodying equalization and the distribution of the braking effort, is met in the proper design of the brake-actuating mechanism. There are four combinations in which couples tending to cause side-skidding are formed, while still another combination tends to cause front-wheel skidding that excises, or reduces, the control of steering. The formation of couples can be prevented by maintaining the braking effort on one side of the car always equal to the braking effort on the other side. The greatest number of exigencies can be taken care of when the brakes are equalized universally; that is, when the front pair is equalized against the rear pair and the brakes of each pair are equalized against each other.

The cause of front-wheel skidding lies in the preponderance of the braking effort of the front brakes over the braking effort of the rear brakes. To eliminate this condition, it is necessary not only to equalize the front pair against the rear pair of brakes but also to adjust the leverages so that the distribution of the applied effort is in proportion to the weight of the front and the rear wheels during braking. The ratio of leverages is determined best by experiment. No method of hand adjustment is precise enough; it is more often dangerous, and no method of equalization depending on elastic deformation is adequate to accommodate all the variables. An automatic means should be provided and this, preferably, is some form of differential mechanism.

In the actuating mechanism shown in Fig. 23 that we have adopted, the front and the rear pairs of brakes are each equalized by the differential action of the continuous cable in the cross-shaft and lever-arms. The cam-lever of one brake is connected to each end of the cable. The cross-shaft is operated by an integral lever. Obviously, the pull at each end of the cable must at all times be equal. The front pair is equalized against the rear pair through the chain. The tension at each end of the chain is the same, while the pull on the rear cross-shaft pulley over which the chain passes has a definite relation to the pull on the chain. For instance, if the rear brakes offered no resistance, the pulley would simply move forward and there would be no tension in the chain; consequently, no effort would be applied to the front brakes.

Facility of Brake Action

Considering the fact that the brakes must act with ease and directness, the choice of a source of power for actuating four-wheel brakes is influenced by two fundamental factors; (*a*) the human element and (*b*) the amount of effort required.

Every normal person reacts physically in the same general way to an external stimulus. Repeated conscious effort, even when moderate, soon causes fatigue or discomfort. Therefore, manual operation of brakes would not meet with general approval if the required effort were too great. On the other hand, human exertion under the excitation of an emergency may be several times greater than even a willful conscious effort and, therefore, a dynamic actuating device such as a servo-mechanism might prove dangerous except for expert usage. With remote control, the nicety of operation is not so good, and it is a very desirable characteristic. Direct manual actuation is safer because of the more simple and direct construction that requires only the ordinary care generally given to two-wheel-brake installations. The fact that it has always been more or less difficult, in the case of heavy cars, to provide sufficient leverage without requiring considerable pedal effort, and to maintain sufficient travel for the brake pedal and still have sufficient reserve travel to take care of the wear of the linings, gave the problem of manual operation of four brakes a questionable aspect until the shortcomings of ordinary linkages were determined.

Mechanics of Brake Construction

The critical analysis of the mechanics of conventional brake construction and operating linkages reveals that

(1) All work done by the force exerted on the pedal when it travels through the distance available is expended in overcoming friction or causing elastic deformation

(2) The overall mechanical advantage is practically constant throughout the pedal travel

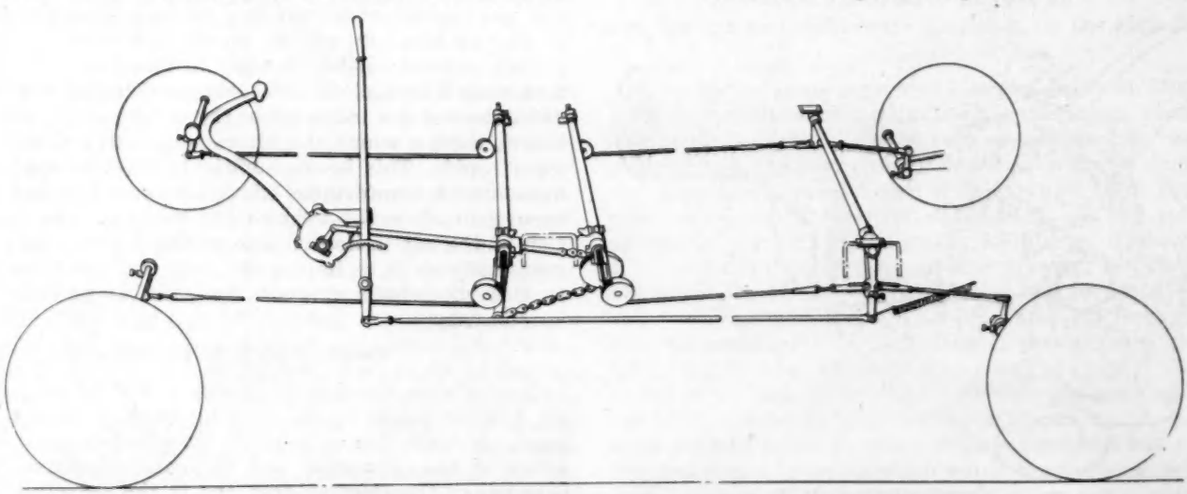

Fig. 23—Diagrammatic Arrangement of the Single-Eight Braking System

(3) Displacements due to deformation of the shoe facings and linkage are considerable

(4) Relatively large clearances must be allowed between the shoe and the drum to insure the complete release of the brakes

(5) Allowances must be made for wear of the brake-shoe facings

In explanation of item (1), suppose that no shoe-clearance were necessary, that by using metal facings on the brake-shoes and absolutely rigid linkages the total displacement of any particle in the system would be zero, and that, since work is the product of force and distance, no work would be done regardless of the forces involved. From this it is evident that a mechanical advantage could be chosen so that tremendous forces at the shoes could be created by a very slight applied force. The problem would then be simple. Displacements that are unavoidable due to extensions, deflections and displacements required for clearance, plus the wear allowances, become serious drawbacks to the use of high mechanical advantages. In fact, the mechanical advantage is limited by these displacements.

This can be illustrated by citing a concrete example. Let

a = the clearance at the middle of the brake-shoe

b = the compression of the shoe-facing plus an amount equivalent to the extension of the connecting linkage at the middle of the shoe

c = an allowance for wear before requiring adjustment at the middle of the shoe

r = the mechanical advantage, foot pedal to brake cam

n = the number of brake-shoes

Then, for a pedal-pad travel of 5 in., we would secure a mechanical advantage of

$$r = 5/n(a + b + c)$$

Thus, for a mechanical advantage of 18, which represents an average value for brakes on the rear wheels only, the equivalent movement that must be allowed for at the center of the brake-shoe is equal to 0.07 in. per brake-shoe; that is, the value of $(a + b + c) = 0.07$ in. If the same allowance were made for four brakes the pressures on the shoe would be about one-half the pressure of two brakes, and the total braking effort would be somewhat less than that of two brakes for a given pedal-pressure. This, of course, is due to the greater friction in the four-brake linkage and to the increased resistance of the larger number of retracting springs. Such a condition would partially defeat the purpose of the four-brake installation and, therefore, a more suitable actuating mechanism is required.

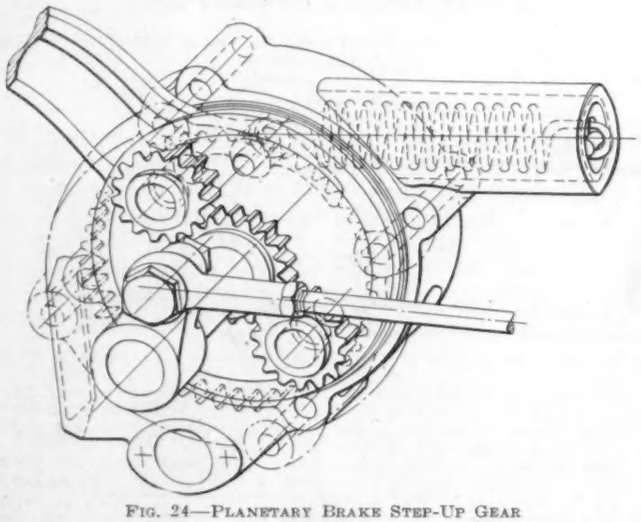

FIG. 24—PLANETARY BRAKE STEP-UP GEAR

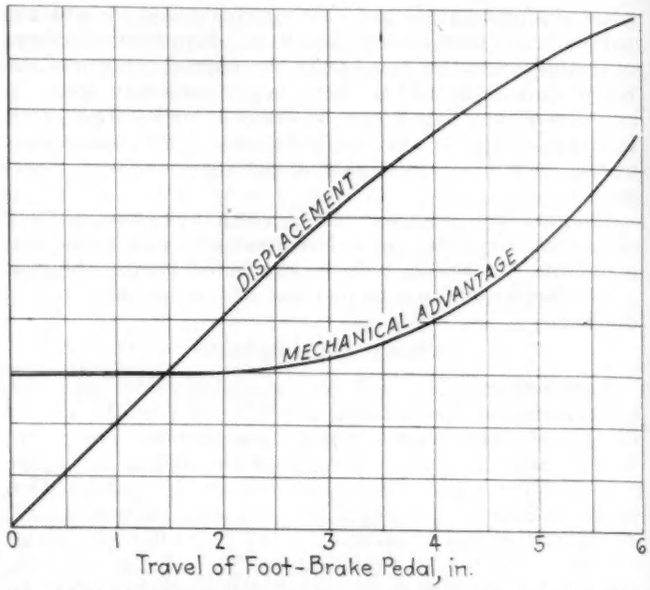

FIG. 25—CURVES OF THE DISPLACEMENT AND MECHANICAL ADVANTAGE CHARACTERISTICS OF THE PACKARD FOUR-WHEEL-BRAKE MECHANISM

The requirements for such a mechanism constitute a varying mechanical advantage such that

(1) The initial clearance of the brakes is taken-up quickly in a small displacement of the brake pedal

(2) No sharp transitions take place; (*a*) making brake adjustment a critical requirement and (*b*) impairing the characteristic "feel" to the driver

(3) There is sufficient latitude so that the operator is warned adequately as to when to adjust brakes

(4) There is a sufficient ratio, but effective braking for a reasonable applied effort

Various types of varying leverage mechanism are available, but all those considered were found to be deficient in some of the requirements that have just been set forth. To meet these requirements, the epicyclic step-up gear was invented. This mechanism is shown in Fig. 24. To illustrate the action of this device, Fig. 25 shows a displacement curve of the brake camshaft-lever movement plotted against the movement of the pedal-pad; the mechanical advantage for various pedal movements is plotted also. It will be noted that both are smooth curves without sudden changes in direction. With such a varying leverage, the clearances and initial stretch and deformations are taken-up in the early part of the pedal travel, during which the leverage is low and the movement rapid. This leaves sufficient travel to operate the brakes with comfortable effort and also provides travel enough to allow for wear of the linings. The deceleration of the car is limited also to the degree that on dry road surfaces it is impossible, with normal adjustment, to lock the wheels even if the pedal is pushed to the floor-boards.

COMFORTABLE STEERING

Comfortable steering is largely a matter of reducing the friction losses in the steering mechanism to a minimum. Friction losses occur in the steering-pivot, in the joints of the cross-tube and the connecting-rod, in the bearings of the steering-gear and in the reduction of the

steering-gear. The fact that these various losses are cumulative makes it important to give careful attention to the design at each of the points where losses occur. When the mechanism is new, these losses are generally lower than they are at any other time during the use of the mechanism.

One of the places of greatest friction loss in front-axle design is at the steering-knuckle pin. Loads on the steering-knuckle-pin bearings are due to a combination of the thrust caused by the weight on the axle and the radial forces due to the beam action of the knuckle. Due to the extremely low rubbing velocities of the surfaces of the bearings, when any displacement of the knuckle occurs it is difficult to maintain efficient lubrication of the surfaces in contact. Further, due to space limitations, the loading is relatively high and, therefore, with bearings that function through the relative sliding of the surfaces, there is usually considerable friction. Friction of bearings, such as ball or roller bearings, that function through the rolling contact of the moving elements, is almost wholly independent of the velocity of the parts. This is particularly true at very low speeds approaching the static condition.

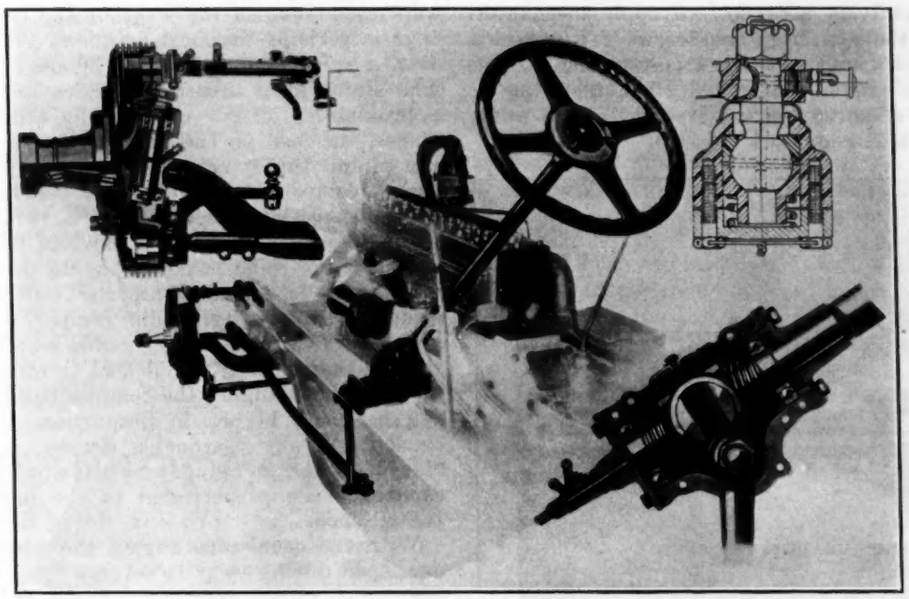

FIG. 26—DETAILS OF THE FRONT-AXLE, STEERING AND BRAKE CONSTRUCTION

AXLE CONSTRUCTION

Fig. 26 shows the construction of the single-eight axle. The lower bearing is a combination radial and thrust bearing while the upper bearing, which is a two-row ball-bearing, takes radial loads only. Very heavy radial loads can be imposed on the lower bearing under conditions such as driving on badly rutted country roads and through side-skidding and steering forces, but these happen with a frequency that is so small in comparison with the frequency of impact in thrust loads that, in determining the load-line of the ball contact, the thrust condition was considered to be most important. Thus, the load-line of the ball contact was made having an obliquity of 30 deg. with the axis of the steering-knuckle pin, and the design of the bearing was worked out so that, under normal static conditions, the maximum stresses produced would not exceed one-fifth of the elastic-limit of the material. On account of space conditions, the double-row ball-bearing is considerably oversize.

The ball joints for the cross-tube are constructed especially to reduce the friction further. In ball joints of conventional construction, it is necessary to make the take-up spring very heavy to enable the forces to be transmitted without too much separation of the spherical seats. The great stiffness of the spring creates a considerable pressure between the surfaces of the ball and its seats and, therefore, produces considerable initial friction. To eliminate this condition, the spherical seats are divided in a plane containing the axis of the cross-tube; hence, the steering forces have little tendency to separate them. Therefore, a comparatively light spring can be used to compensate for wear.

STEERING-GEAR EFFICIENCY

Steering-gear efficiency is largely a matter of design for the screw-and-nut type. The efficiency for motion opposed to the direction of the force acting on the nut is given by the formula

$$e = \tan \gamma / \tan (\gamma + \delta)$$

and, for motion in the same direction, by the formula

$$e = \tan (\gamma - \delta) / \tan \delta$$

where γ is the lead angle and δ is the angle of friction.

Thus, the efficiency approaches the maximum when the lead angle is 45 deg. Typical helix angles used for worm-and-nut steering-gears range down as low at 6 deg. In our design, we have chosen a helix angle of 8 deg. 30 min. which has a steering efficiency of 58 per cent as against 25 per cent for an angle of 6½ deg. The amount of irreversibility, however, is also greater; for the 6½-deg. angle, the efficieny is 7.6 per cent and for the 8½-deg. angle it is 29 per cent. This computed efficiency, however, is based on ideal conditions of the surfaces in contact; that is, upon efficient lubrication.

The conditions under which the steering-gear operates are particularly unfavorable to lubrication. A greasy layer formed by oil or other viscous lubricant has a very low resistance to penetration when the rubbing velocity is very low and, if the pressure between the surfaces is

sufficient, the oil-film is ruptured and the coefficient of friction is increased greatly. The steering-gear, under normal driving, is subject to impact loads of considerable magnitude. These, of course, are transmitted through the threads in contact. If the threads bear only in spots, the pressure becomes very great over the small areas; hence, lubrication is impaired by the rupture of the lubricating film, or solid friction obtains and the gear is then difficult to operate. It is practically impossible to machine a screw and nut to fit with all the thread surfaces in contact and yet make them fit so snugly that there is no end-play.

To insure effective lubrication and to eliminate end-play, which is undesirable for many reasons, we have adopted the following construction: The screw is first made very carefully to close limits of accuracy, especially for the lead. The threads in the nut are then cast to each individual screw, the threads and the backing forming an annular lining in the bronze shell that forms the outer section of the nut. After the babbitt cools, the threads fit so snugly that it is necessary to apply mechanical power to loosen the nut from the screw. To obtain a free working fit with sufficient clearance for good lubrication, the threads are lapped in oil. The resulting fit shows a uniform effective bearing over the entire working surface of the thread.

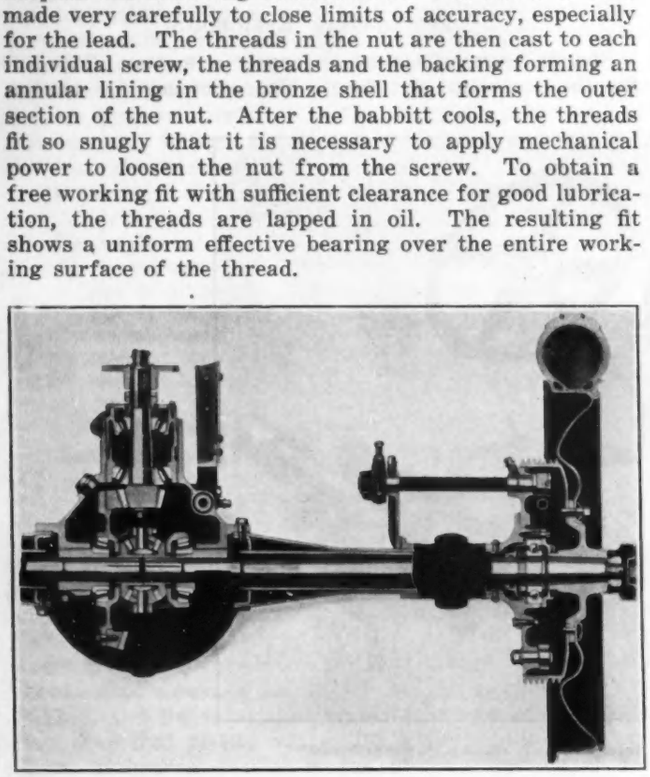

FIG. 27—DETAILS OF THE REAR-AXLE AND BRAKE CONSTRUCTION

THE REAR AXLE

Economy, as we have seen, implies dependability and durability; for, without securing these in a very large degree, there is necessarily an expenditure of time and money. Dependability and durability in rear-axle construction do not refer only to uninterrupted performance of the axle as a means of transmitting power; that it not sufficient, for the element of taste enters into the consideration. Few things are more irritating than a noisy rear axle and, naturally, the owner does not buy his new car when it has a noisy axle; the noise develops later. But the owner who continues to drive his car very long after the axle develops noise is not above criticism; for, in most cases, if immediate attention is given, expensive repairs and replacements can be avoided as well as the needless aggravation.

The development of axle noise is a useful warning that something has gone wrong and indicates that one or several conditions prevail such as gears being out of adjustment, worn bearings, a broken or distorted bearing, poor lubrication, foreign substance in the axle, worn gears or some developed structural weakness of the axle. These, of course, are not the only causes for noise in axles and do not include those met in the design and manufacturing processes; they are only the fundamental ones in cases where the noise is due inherently to a change from previously satisfactory conditions. Frequently, axle noise is produced by the use of certain types of tire, by the use of tire chains or by other indirect causes, but, in such cases, the axle is not to blame. If the car is driven with the axle in any abnormal condition, the noise will become worse, to say the least, and it is necessary eventually to visit the service-station; but generally something more serious happens, a break is almost inevitable and breaks happen frequently at the most inopportune times and are expensive. However, having recognized the causes of the development of axle noise, it is the engineer's work to eliminate them if possible; or, if not possible, to reduce their effect.

Of the various causes for the development of noise or the subsequent more serious troubles, the change in gear adjustment due to the wear or the failure of the bearings is perhaps the most frequent. In spiral-bevel-gear sets, a spiral angle of about 30 deg. is used generally. The pinion axial thrust for angles in this neighborhood approximates 50 per cent of the tangential tooth-load. Hence, the load on the bearing that is reacting against the pinion thrust comprises a radial component and a thrust component of nearly 50 per cent of the radial component; the direction of the resultant load-line is inclined about 60 deg. to the axis of the bearing.

With taper-roller bearings having the small cone angles common to usual designs, the complementary thrust component for a given radial component is much smaller and, therefore, a secondary radial load that acts in addition to the imposed radial load is created. By the use of larger cone angles, the complementary thrust capacity can be made higher in proportion to the radial load capacity to any reasonable degree, the maximum efficiency of design being reached when these component capacities are proportional to the imposed thrust and radial loads.

With the usual cone angles, the amount of axial play developed during wear is several times greater than the amount of radial wear and, therefore, the gear adjustment is affected quickly. The proportion of end-play that is developed with a similar amount of wear of the bearings in which the cone angles are larger, is greatly reduced and thereby maintains the gear adjustment for a much longer period for a given amount of wear. These two advantages of steep cone-angles are especially important in large high-powered cars in which the axle design is necessarily more compact than in lighter cars of less power. Fig. 27 shows the layout of the single-eight axle in which these bearings have been adopted.

In the case of the differential bearings, an increase in the diameter of the outer race results in a lower resistance to distortion from the true circular form. It is important that the circular elements be maintained; as otherwise, secondary stresses will be set-up during the operation of the bearings. For this reason, wherever possible, it is a great advantage if the strength of the bearing races are aided by their mountings in resisting the deformations caused by the imposed loads.

Where the differential bearings become large, as in the case of axles for cars of the single-eight weight and power, the usual construction of split differential mountings in which the outer race of the bearings is clamped under a cap possesses the fundamental disadvantage of a lack of rigidity. In the first place, very great care

must be exercised in the machining operations to insure an almost perfect fit between the race diameter and the bore into which it is to fit. This is initially difficult, as the cap is bolted in place in the differential carrier and the bore is ground while the cap is under an entirely different system of stresses than those for the condition in which the bearing is clamped in place. After the grinding operation is finished and the bearing mounted in place and clamped, the circular form of the bore is no longer maintained. The result of this is that the bearing is subjected in operation to localized stresses wherever the diameter is the smallest, although there is a tendency for an elastic adjustment to take place in the bearing.

Generally these localized stresses are very much higher than the normal stresses for which a better distribution of the loading of the rollers and the races is found. Bearings operating under these abnormal conditions show two characteristic types of failure; the races become badly pitted in the areas of stress localization, or the race cracks completely. Frequently, however, rollers show failure; but, in this case, it is difficult to determine whether the failure is caused by race distortion or because of improper adjustment, unless the races themselves show distress. To add to the distortion caused in the original clamping of the bearings, the cap itself is subjected to a stress distribution that is not favorable to the maintenance of a circular form under the applied bearing loads.

To eliminate these fundamental disadvantages in the mounting of large bearings, we have adopted the construction as shown in Fig. 27, which is a mounting and type just the reverse of the clamp mounting. In this construction, the outer race of the bearing is pressed into the solid circular annulus of the differential case. With this construction, the circular form of the outer race is maintained accurately. The form of the inner race is also maintained in this mounting. The inner race is pressed-on to the end of a carefully ground cylindrical sleeve. This sleeve is stepped in two diameters, the larger of which is relatively long and fits a closely reamed seat in the differential carrier. The outer end of the sleeves is threaded for adjustment. Such a mounting eliminates the formation of any secondary stresses and thus allows the bearing to do its work most efficiently and have the longest life.

Conclusion

I have pointed out that the basis of motor-car transportation comprises the elements of performance, safety, economy, comfort and taste. To these every thought and effort devoted to the engineering and production of the modern automobile will find a definite causal relation. We strived for a certain degree of performance in coordination with the other elements and achieved that which we set out to do. We determined on a degree of safety for large cars that we felt, in addition to the standard of inbuilt safety that we have always tried to maintain and the improvement in steering control which we accomplished, required a brake on each road wheel. The simplicity and accessibility of the engine, along with the improvements that are constantly being effected in the advance of craftsmanship in engine building; the improvement in the transmission and rear-axle design; and the improvement in many minor details have, we feel, established a new standard in economy above that reached by our previous efforts on cars of this class. Comfort and taste have, we feel, been satisfied in a larger measure. Approval of the inherent smoothness of the single-eight, to say nothing of that relating to the appearance, the quality and the texture of the upholstery and the many other appeals that have been a constant object of our efforts for years, we leave to the user's own judgment.

THE PRODUCTION AND USE OF SILVER

SILVER is produced largely as a by-product, since the ores of which silver is a derivative are mined primarily for other metal content, gold, copper, lead and zinc. Reputable estimates state that silver production comes 70 per cent from base metal ores; 30 per cent from precious metal ores. The amount of silver found is of secondary consideration. Therefore, the world's supply of the white metal is involved with many cross currents of the supply and demand for other things.

According to figures prepared by the Director of the Mint, the annual silver production of the world was running about 225,000,000 oz. in 1913, just before the war began. Since then it has not reached as much as 200,000,000; the low point was touched with a production of only 161,000,000 oz. However, a production of 198,000,000 was reached in 1918 when the war demanded large supplies of metals generally. The figures show that as a rule the North American Continent contributes about three-quarters of the annual world production; the United States usually rather more than one-third. Mexico, before the revolution, ranked first as a silver mining country. But for a long time thereafter production was but a fraction of that of former years. Indeed, the curtailment of world supply from 1914 through 1921 can be largely traced to the Mexican revolution. Given a continued stable political condition in Mexico, it is thought that production there may hold the pre-revolution stride which it picked up again in 1922. Last year the 81,000,000 oz. produced broke production records for recent years. Some of the Mexican mines have been operated for hundreds of years, but experts think that silver deposits show no signs as yet of exhaustion.

After Mexico in silver production comes the United States; indeed, during many of the revolutionary years the United States was ahead of Mexico. The third place belongs to Canada; the remaining quarter of the world's annual production is divided among the other continents.

North America produces the surplus of silver; India and China absorb it. This generalization appears to be fairly well substantiated in experience. When the affairs of the world were flowing along relatively evenly in the years just before the war, there was a rough equation between the exports of silver from the United States and the imports of silver into India and China. During the years 1910 to 1914, the United States annually exported about 100,000,000 oz.; the annual net import of China and India was about 90,000,000 oz. It is perhaps fair to consider the exports of the United States as representing the surplus product of North America, as most of the exports of silver of Mexico and Canada leave North America through the ports of New York and San Francisco. With North America balanced against the Orient, the silver produced by the rest of the world supplies the demand for silver in the rest of the world, roughly speaking.

The war created a tremendous demand for the products of India, cotton, jute, rice, wheat. And at the same time it cut down the return flow of imports from the rest of the world, piece goods, iron and steel. To strike a balance of trade with India, the precious metals were much in demand. The gold of the world found a particular magnet in the United States; India was the magnet for silver. Hand-to-hand coins, note reserves, personal ornaments, individual hoardings in India have always been chiefly silver. In the war years the silver that the rest of the world paid for Indian exports was largely buried in that land where every man is his own banker.—*Commerce Monthly.*

Engineering Brains in Fleet Operation

By E. E. La Schum[1]

Metropolitan Section Paper

THE magnitude of the business of the American Railway Express Co. requires that careful consideration be given to the details necessary for economical operation. The equipment comprises 12,755 vehicles, of which approximately one-third are motor driven and have a carrying-capacity of more than one-half the total. On July 1, 1918, when all the express companies were merged into one organization, it was found that the motor-vehicle equipment included 59 different makes and 131 different models. Among the 377 trucks built by one company were 21 different models. Diversity of equipment, of course, complicates the maintenance problem and adds to the cost. Additional expense is incurred frequently by purchasing and experimenting with parts offered by makers of accessories such as carbureters, spark-plugs, wheels and the like. Careful inspection, adequate lubrication and the adoption of "stitch-in-time" methods will save needless expense.

Exclusive of drivers' wages, depreciation, interest and insurance, the operating costs are shown to be distributed according to the following percentages: Gasoline, 30.00; cylinder oil, 2.25; tire costs, 2.50; painting, 1.30; body repairs, 4.75; chassis repairs, 35.20; and garage expenses, 24.00 per cent. In the eight largest cities of the Country, 6254 calls for assistance were received at the garages during the year; of these, spark-plug troubles comprised 5.9 per cent; ignition system, 9.1; carbureter, 8.6; gasoline lines clogged, 7.6; broken fans or fan-belts, 9.3; and steering-gears, clutches, transmissions, drive chains and tires about 3 per cent each. In the same cities, the calls for garage service by 722 electric trucks were less than two per truck per year. Exhausted batteries caused 19.1 per cent of the trouble; controllers, 12.1; steering-gear, 6.2; brakes, 5.0; drive chains, 11.6; field or armature burned out, 4.8; and springs, 1.3 per cent.

THE business of the American Railway Express Co. is transportation. In fact, transportation is its sole function. Sight is sometimes lost of the fact that high efficiency in the business of transportation can be attained only through a most careful analysis and a careful consideration of the fundamental problems that necessarily must be solved to assure perfect and economical operation for the longest period of time. Transportation in the case of an express company must be reduced to a science. No sentimental selection of equipment, guesswork method of operation or slipshod method of maintenance can be tolerated. During the year 1922, more than 184,000,000 shipments were handled, and these shipments had to be handled at least once at the point of origin and at least once at the point of destination, in addition to handlings on the way. The average weight per shipment was approximately 82 lb., producing a gross revenue of approximately $294,000,000. From the magnitude of the business it can be seen readily that it is necessary to make a most careful selection of vehicle equipment.

In New York City, we require 655 motor and approximately 900 horse-drawn vehicles. In Chicago, we have in daily service about 648 horse-drawn and 306 power vehicles, including 25 tractors with about 60 trailers. Our total vehicle equipment throughout the United States and Canada consists of 2636 gasoline-driven vehicles, 1195 electric street-trucks, 324 electric industrial platform-trucks and 100 semi-trailers and about 8500 horse-drawn vehicles. This is a total of 12,755 units of which approximately 33 1/3 per cent are motor vehicles; but the 33 1/3 per cent of motor vehicles, to the best of my knowledge, has a carrying capacity of more than 50 per cent of the total. Statistics show that, in our service, the horse-drawn vehicle averages approximately 12 miles per day, the electric vehicle 20 miles per day and the gasoline vehicle 30 miles per day, which surely is sufficient to prove that, if we could get a record of ton-miles, the motor vehicles are doing by far the greatest amount of actual work.

In the beginning, from 1912 to about 1917, the experiment was in the making and there was need of engineering brains in the maintenance end, because the changes were so rapid that only alert quick-thinking maintenance men could hold the pace set by the engineers. It is a lamentable fact that on July 1, 1918, when all the express companies were merged into one, the new organization found itself in control of 59 different makes of motor vehicle and 131 different models. There was a large number of old-line, or so-called manufactured, trucks. One would think that if we had purchased only one make we should have had a standardized maintenance proposition but, out of 377 trucks built by one old-line company, we had 21 different models in which the parts were interchangeable only in a small degree, and one can see how far from standardization we were kept by the engineers. In the trucks recently bought from an old-line company, changes were made without notice to us, after we had received the first lot and before we received the second lot, even though the second lot was purchased within a few months after the first, so that it was necessary for us to carry two types of radiator in the stockroom when one should have been enough. In one particular case, in a fleet of eight trucks received from an old-line company, all delivered at the same time, there were three different makes of magneto. This great difference in models and in constructional units complicates our maintenance problem and adds much to its cost. We realized, though, in the beginning, that we were starting with an industry in which there would be rapid development and, while we were saddened by the cost of maintenance, we are gladdened by the fact that the developing of motor trucks and the standardizing of parts has, we believe, reached a stage of common practice. In selecting equipment, our engineering knowledge teaches us that the words "manufactured" and "assembled" are dissolved into one word and that a so-called manufactured truck and a so-called assembled truck in which the important constructional units have been designed properly and properly assembled will be found to be identical, and that with either truck maintenance should be performed economically and the satisfactory performance of the truck be assured.

The selecting of motor-vehicle equipment certainly has much to do with the maintenance problem, and here again engineering knowledge is necessary to keep away from

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complicated apparatus or apparatus that is vastly different from standard or common practice. The fleet-owner or the buyer of motor trucks should not complain or criticize engineers for what has gone before. It was necessary to study, work out and improve the designs, but though engineers are able to produce a motor vehicle that will perform satisfactorily and economically, there is still need of engineering brains to avoid the expense of hundreds of accessories offered by others than the truck builders. We receive trucks equipped with carbureters that were worked-out by engineers, yet hundreds of carbureters are changed daily because the owners or the buyers have been influenced by sales talk. I do not know how many spark-plugs there are to be had, but I do know that there is only a slight difference, if any, in their intrinsic value, and that the owner or the operator who is continually trying different spark-plugs is creating an unwarranted expense.

Necessity for Engineering Knowledge

I believe it is unnecessary to have an engineering staff to supervise the actual maintenance and repairs, but I believe it is necessary to have engineering knowledge to keep the mechanical forces or inexperienced executives from substituting or changing from one thing to another that will prove no better. Take, for instance, the subject of wheels. The old wood artillery-type wheel has given comparatively little trouble; the solid steel wheel may be just as good; in fact, I think it is, and I believe there is a real benefit to be derived from the use of the so-called resilient wheel under certain conditions, but truck owners should not be influenced to the extent of incurring an expense for any accessory unless they have sufficient indisputable knowledge that the expense will be returned by economical performance or extended life. Therefore, before changing from the conventional type of wheel or the conventional type of tire and adopting some of the so-called resilient wheels or semi-pneumatic tires, a careful analysis should be made by a competent engineer whose unbiased mind should enable him to advise what is best.

In the old days, the frames were a continual source of trouble but, under the practice of today, it takes something out of the ordinary to break them. We do not have broken frames to contend with in regular service, but we should have them if we did not exercise proper supervision over operation and, by inspection and stitch-in-time methods, keep the rivets tight, the wearing parts properly lubricated and the natural wear corrected. The same thing applies generally to most of the other principal units of the truck; that is, to the steering, transmission and axle units, which should be lubricated and adjusted to compensate for the natural wear. There is hardly any reason for neglect, other than gross carelessness, when matters need attention on units that afford easy access for inspection and repair. Occasionally, faults develop in the hidden mechanism of the engine and, due to poor judgment, they are not given attention in time to prevent expensive repairs. The engine is deserving of and should have careful inspection by competent repairmen so that it may have a fair opportunity to operate satisfactorily with the other units of a well-designed and well-built motor truck for a reasonable period, until general repair or rebuilding becomes necessary. We know that much has been accomplished by engineers in the way of simplification and standardization and we should like to see them continue, were it not for the fact that carrying standardization too far in motor vehicles would deprive them of individuality. To my mind, building motor vehicles has reached the stage of common practice, but the engineers must engineer some individuality into the cars.

TABLE 1—OPERATING-COST DISTRIBUTION

Item	Per Cent
Gasoline	30.00
Cylinder Oil	2.25
Tire Costs	2.50
Painting	1.30
Body Repairs	4.75
Chassis Repairs	35.20
Garage Expenses	24.00
Total	100.00

I have asked engineers to develop a mileage-counter, but there seems to be great indifference on the part of truck engineers, who have let the old hub-odometers remain. This instrument probably would record regularly and accurately were it not for the fact that it is placed in the most hazardous position possible. The difficulty with it is that it will not withstand the abuse of collisions with curbstones and elevated railroad pillars and with other hubs. It is impossible for the truck owner to keep records based on actual miles traveled, because no hub-odometer will withstand such usage. Also, I have asked engineers to provide a standardized radiator-guard or at least to complete the work when they turn it out by applying a radiator-guard. In many cases, they leave this to the owner. I have tried repeatedly to find out why it is necessary to remove the seat cushion in a motor truck, almost without exception, to fill the gasoline tank. Why can we not have a filler-pipe extending through the seat panel so that the tank can be filled without disturbing the seat cushions and the cab curtains?

Distribution of Maintenance Costs

The figures in Table 1 will be interesting to those who have not gone into operating-cost distribution in detail; the figures do not include drivers' wages, depreciation, interest and insurance.

Gasoline is one of the most important items of expense, and a live garage superintendent will always watch that item. For some reason, gasoline seems to be the common prey of everybody. It is more likely to be used indiscriminately than any other item. Employees use it for washing parts, and employees and officials of the firm using their own cars for the company's business feel that they should have their gasoline tanks filled; but all these things can be and no doubt are controlled. Gasoline should be saved, first, by knowing that the carbureter is right for a particular truck, and that if any adjustments are to be made they should be made by a capable carbureter man. Better still, a non-adjustable carbureter should be used, if possible. The drivers should be taught that a rich mixture, which may get them over the grades without shifting the gears, is bound to waste 35 per cent; this waste will accumulate carbon which, within a short time, will cause the engine to work badly. As a matter of fact, if a driver would shift the gears, he could go over the grade with the machine in better condition than a person can who believes and relies upon the wonderful flexibility or pulling power of his particular truck or passenger car. Numerous gasoline savers or adulterants are offered for sale. The vendors claim to eliminate carbon and to crack or break-up the gasoline in such a manner that much greater mileage will result. I have tried a number of these gasoline adulterants without any benefit, and I believe that the Society will bear me out in the statement that, from

the standpoint of saving gasoline, so-called gasoline savers of the liquid, powder, crystal or tablet type are not valuable. I believe that auxiliary air devices that are operated from the dash as the driver may see fit will accomplish nothing; but there is a possibility that an auxiliary air device may be constructed so that it will function only at pre-arranged engine speeds, by hooking it up with an accelerator pedal or with a fly-ball governor. Then again, this means only a lean mixture and, to go back to what I have said before, if you can teach the driver to use the gearset installed by the builder of the vehicle, at least 35 per cent of the gasoline now used can be saved.

Nothing connected with the maintenance of motor vehicles can be more confusing to the lay mind than the selecting of cylinder oil suitable for lubricating engines. I believe that oil refiners are serving the Country with the best that can be obtained from crude petroleum, but the owner or the buyer generally knows little about oil and must make a selection from hundreds of samples.

The next item of expense is tire cost, which we must agree is very low. As a matter of fact, tire cost today is approximately 25 per cent of what it was in 1913 and 1914. Solid and pneumatic tires are showing at least four times the mileage they did in the period mentioned, and they are giving from two and one-half to three times as much service as the old mileage guarantee, which was from 3500 to 5000 miles on pneumatic tires and from 7000 to 8000 miles on solid tires. The percentage of replacements for defective workmanship or material is so low that it is of practically no importance.

The prices of painting, of course, will differ, but, generally, to keep trucks well painted the cost should not exceed that which I have mentioned.

Bodies with durability built into them in the beginning should not cost more than the amount given for repairs, and I believe they should last fully as long as any chassis upon which they have been installed.

The greatest item of expense is mechanical repairs to the chassis. That surely should not exceed the amount I have mentioned, and here again I fully believe the cost can be kept down only by taking a stitch in time; by lubrication, which it is criminal to neglect; and last, but by no means least, by instructions worked out to give the driver a more extended conception of his duties and of his responsibility for the valuable equipment placed in his hands. The distribution of chassis-overhauling costs is stated in Table 2.

Garage expenses can be controlled because, generally, they are fixed, with the exception of light and heat. As a matter of fact, light and heat are controlled by the seasons, to a great extent.

Analysis of Emergency Calls

For the year 1922, I have taken the cities of Atlanta, Buffalo, Chicago, Cleveland, Detroit, New York, San Francisco and Washington. In these cities you have the snow belt, cities that have little snow, and cities that have no snow and no cold weather. In these cities there are 802 gasoline trucks, very few being of less than 2-ton capacity, the greatest number being 2 and 3½-ton capacity and a considerable number being of 5-ton capacity. Generally, there are 300 working days per year, but, from these 802 trucks, we have obtained 306,348 working days of 8 hr. each. The total mileage, as near as we know, was 9,190,440 miles.

During the year we received 6254 calls for help from these trucks; something was the matter with the truck, and the driver called for assistance from the garage.

TABLE 2—DISTRIBUTION OF CHASSIS-OVERHAUL COSTS

	Labor Time		Labor Cost		Material Cost		Overhaul Cost	
	Total, Hr.	Per Cent	Total	Per Cent	Total	Per Cent	Total	Per Cent
Dismantling	67.42	9.27	$50.57	7.21	$0.84	0.12	$51.41	3.65
Engine	153.42	21.09	156.84	22.38	397.85	56.16	554.69	39.33
Carbureter	8.17	1.12	8.39	1.21	0.60	0.08	8.99	0.64
Magneto	4.00	0.55	4.17	0.59	2.43	0.34	6.60	0.47
Radiator, Muffler	15.25	2.10	12.00	1.71	0.56	0.08	12.56	0.89
Clutch	7.17	0.99	7.30	1.04	14.44	2.04	21.74	1.54
Transmission	36.08	4.96	37.53	5.35	51.68	7.28	89.21	6.33
Differential	19.75	2.72	20.28	2.89	46.22	6.51	66.50	4.72
Jackshaft	3.50	0.48	3.52	0.50	.....	.....	3.52	0.25
Frame	62.58	8.60	62.44	8.91	1.11	0.16	63.55	4.51
Frame and Radiator Construction	24.83	3.41	25.42	3.63	44.27	6.25	69.69	4.94
Steering-Gear	41.50	5.71	43.07	6.14	1.78	0.25	44.85	3.18
Brakes	34.83	4.79	36.13	5.15	28.83	4.07	64.96	4.61
Radius and Tie-Rods	7.83	1.08	8.14	1.16	5.95	0.84	14.09	1.00
Springs	18.95	2.61	19.23	2.74	15.17	2.14	34.40	2.44
Wheels	.....	.....	.....	.....	5.63	0.79	5.63	0.40
General Assembling	216.50	29.76	200.48	28.59	0.95	0.13	201.43	14.29
Miscellaneous	5.50	0.76	5.58	0.80	90.50	12.76	96.08	6.81
Grand Total	727.28	100.00	$701.09	100.00	$708.81	100.00	$1,409.90	100.00

This was a little less than seven times per truck per year. An analysis of the causes of these calls, stated in percentages of the total number, is made in Table 3.

There were only 50 cases of bearing-trouble, and rarely a case in which a full set of connecting-rod bearings burned out. The spark-plug trouble in Table 3 was due, almost without exception, to the fact that the driver had been using a choker and had fouled the plugs by too rich a mixture; this gave us an opportunity to teach him that the trouble could have been avoided if he had used the gearset. During the same period and for the same number of trucks the spark-plug replacements on four-cylinder engines have been a little less than six per truck per year, or a little less than one-half of one spark-plug per truck per month. The ignition system parts other than the spark-plugs, such as the magneto, the distributor, the wiring and the switches, that caused the trouble stated in Table 3, required only slight adjustment or repair as a rule, because the trouble was of minor importance. This work was done at once and the truck was able to resume its work immediately. The miscellaneous troubles in Table 3 were minor, such as

TABLE 3—ANALYSIS OF CAUSES OF TROUBLE CALLS

Cause of Call	Per Cent
Engine Bearings	0.8
Spark-Plugs	5.9
Ignition System	9.1
Carbureter	8.6
Gasoline Lines Clogged or Leaking	7.6
Broken Fans or Fan-Belts	9.3
Radiators	1.1
Water Pump or Water Line	1.6
Radiators or Pumps Frozen	0.2
Governors	0.5
Steering-Gears	3.3
Clutch	3.0
Propeller-Shaft	1.5
Universal-Joints	1.1
Transmissions	3.0
Differentials	0.5
Rear Axles	0.9
Radius-Rods	0.9
Drive Chains	3.8
Wheels	1.5
Springs	1.8
Brakes	2.2
Tires	3.3
Lights	1.0
Accidents, only	1.6
Out of Gasoline or Oil	3.1
Miscellaneous	23.8
Total	100.0

the inability of the driver to start the engine, or on account of the stalling of a truck in a ditch or in the mud or snow.

During 1922 and in the same cities, we had 722 electric trucks; from these we obtained 226,716 working days. We were called out less than twice per truck per year on account of these trucks, for a total of 4,987,752 miles. During the year we had approximately 91 electric motors burned out; that is, either the field or the armature burned out. This was 4.8 per cent of the trouble, but it is small when the total mileage is considered. In many cases it was due directly to the fact that the trucks encountered deep snow, mud, ice, cold, rain, heat and, in all probability, considerable overloading. We had 366 cases of exhausted batteries, which caused 19.1 per cent of the trouble, or an equivalent of one case of trouble per truck in 2 years. The other items of importance were controllers, which caused 12.1 per cent; steering-gear, 6.2 per cent; axles, 1.5 per cent; brakes, 5 per cent; springs, 1.3 per cent; and drive chains, 11.6 per cent. On the modern electric truck drive-chain troubles have been eliminated. The remainder is made up of miscellaneous items too numerous to mention, but those who are doubtful of the ability of the electric truck to perform should take note of the fact that 722 electric trucks performed better than 314 truck-days per year, with less than two trouble calls per truck per year. Surely, all of the above is an indication of the reliability of the motor vehicle and its component parts and an indication of the service that can be accomplished under proper management.

The business spotlight is being turned on the motor truck and the organization back of it. The light should be focused first on the organization of maintenance; second, on the organization of operation. Each organization should be required to analyze costs and to study legislation referring to truck operation. Each organization should be required to analyze costs in order that nothing shall be left undone that will bring about greater efficiency and satisfaction to the owner.

THE DISCUSSION

J. F. WINCHESTER:—Engineering brains have conceived the motor truck, have placed it in its present position of reliability and are about to create for it a more varied and a wider field. Some standardized form of cost analysis is needed, so that one operator can compare the results he is obtaining with those obtained by another. Although, in the past, we have had certain standardized forms of bookkeeping to accomplish this, my observation is that they are complicated and tend to result in confusion when direct comparisons are made. Therefore, I suggest that careful consideration be given to the basis on which a cost analysis should be made.

The unit of cost will vary in different industries but, when a direct comparison of the costs of mechanical operation is to be made, should this be by the individual truck from the date of installation, by fleet operation, that is, an average by make and model, or by year; and to what degree or extent of detail should these figures be carried out? Simplicity should be the guide in deciding this question, for there is no doubt that there is a great tendency to analyze the various operating costs in too great detail, which results finally in confusion to a prospective user.

My reason for suggesting a standardized form of bookkeeping is that, in some particulars, the results of my experience do not conform to those of Mr. La Schum and of other authors. Mr. La Schum states that the light should be focused first on maintenance and second on operation. My experience has been, even though I am a mechanical man first, that this order should be reversed. The reason is that the organization of operation controls the items of expense that develop fixed charges, which constitute approximately 65.2 per cent of the total operating charge, while the organization of maintenance controls the group of charges that make up the variable costs, which are largely mechanical expenses, and which my analysis shows constitute approximately 34.8 per cent of the total. My division of fixed and variable charges is approximately as stated in Table 4.

TABLE 4—DIVISION OF FIXED AND VARIABLE CHARGES

	Items	Per Cent	Per Cent
Fixed	Salaries, Chauffeur and Helper	43.4	65.2
	Storage and Insurance	2.0	
	License	1.0	
	Depreciation	18.8	
Variable	Gasoline	11.3	34.8
	Lubricating Oil and Grease	0.8	
	Tires	9.9	
	Road and Miscellaneous Expense	10.4	

In considering unit costs, whether they be costs per ton-mile, per package or per some other unit of value, it is a fact that careful planning, routing and proper equipment in connection with the installation will be the greatest factors in keeping costs down. The variable charges, although constituting 34.8 per cent of the total, include such items as gasoline, lubricating oil and grease, which, provided the equipment is operated efficiently, are difficult to offset. We know that the efficiency of a truck depends to a large extent on individual mechanical design and on materials and that no great economies can be obtained in similar models, provided the truck is watched carefully from an inspection standpoint. I submit, therefore, that the responsibility of the organization of maintenance is secondary to the organization of operation, and that it should render to the organization of operation efficient service from an engineering and a general operating standpoint so that unit costs can be kept down to the minimum figure. Friction between the two organizations should be avoided and harmony should be maintained. Either should be able to pass on the figures of operation, so that constructive criticism can be obtained from both units of the organization.

COMBINED ENGINEERING AND BUSINESS ABILITY

The question as to whether engineering brains should direct the operating of a truck fleet will always be debatable. In supervising either kind of expenditure shown in Table 4, I believe that a business man, no matter how keen he may be, works under a handicap unless he has had some engineering training. The wider his engineering training is, the better position he will be in to discuss and to pass on the many problems that arise.

The points brought out in Mr. La Schum's paper that illustrate problems in connection with carbureters, sparkplugs, tires, wheels, frames, gasoline savers, auxiliary air-devices and the like indicate that engineering knowledge is necessary if these types of accessory are to be passed upon individually, without expensive experimental work. Repair and maintenance work are primarily

mechanical and, if the management is to be efficient and mistakes are to be avoided, the person in charge should be able to judge whether the recommendations made to him are right or wrong from an engineering standpoint. The point to be avoided in any operation is the duplication of experimental engineering work previously conducted by a factory organization, by the Society or by others, the results of which are accessible. I have failed to encounter some of the problems cited by Mr. La Schum. For instance, in the last 2 or 3 years, there have been installed under my supervision approximately 500 trucks, having uniformity in their specifications covering speedometers, magnetos, lighting equipment and accessories, such as batteries, switches, headlamps, tail-lamps and so on, which tend to simplify the service problem and keep down the cost of maintenance after the vehicle is in operation. No difficulty of any importance was encountered with the builders in getting them to use standard accessories.

I have found no difficulty in recording mileage. All mileage reports in the records to which I have access and from the vehicles under my supervision are based on speedometer readings; I believe that at least 98 per cent of the mileage turned in is based on the accurate results obtained from these instruments. A few years ago satisfactory speedometers and driving mechanism were difficult to obtain, but the difficulty was caused by driving the speedometer from the front wheel; that is a bad location, as the instrument is exposed to the elements, and results in the rapid wearing out of the universal-joints and the gears. By developing a drive taken from the rear of the gearbox, this trouble practically has been eliminated. It is also my belief that, within the last several years, front-wheel speedometer-drives have been developed to the point where little difficulty is being encountered, comparatively speaking. By fastening the parts in such a way that the effects of vibration are overcome, the upkeep of this type of drive has been reduced to a minimum.

Attention has been called to the mechanical changes in various models of truck bought from an old-line company. A suggestion that I offer to the Society contemplates the working-up of a mechanical-installation sheet along standardized lines that would give complete mechanical specifications to the truck operator concerning the type of installation he is to make. The sheets, if prepared carefully by the builder of the truck, would record the changes made in its construction and would assist the fleet operator materially in ordering material for a given truck, The sheets should be designed so that they will allow the fleet operator to record standard changes, such as boring oversized cylinders, gear-ratio changes and the like, which sometimes are necessary after a truck has been installed and which, to give satisfactory service in the field, are necessary for the maintenance organization to have ready at all times.

Standard catalogs, as supplied with cars in the past, do not take care of this problem satisfactorily. If fleets are to be handled more economically in the future, it is necessary to have detailed information rather than the general information contained in "parts" catalogs.

Another suggestion is the need of greater speed on the part of the truck industry in supplying fleet operators with complete engineering data regarding a given vehicle, so that a company operating a particular type of equipment can have definite engineering recommendations available regarding tolerance of fits in installing various parts, such as pistons, bearings and the like, then, when overhauling work is being done, there will be a definite assurance that the unit will operate satisfactorily when put together. This, together with a basis for fleet-repair rates, is to my mind a great need in the truck industry if fleets are to be operated economically. Many of the larger operators are able to solve these problems themselves because they produce economical operation; but, in the case of the smaller fleet-operator, this work constitutes an expense that is difficult for him to absorb and hardly justifies his going into.

Fleet Operation Costs

I have never been able to pass-up the subject of tire cost from the standpoint of tire cost only. My experience proves that the type of tire employed has a definite relation to the efficient operation of the truck itself. We in the automotive industry have heard certain claims made by tire manufacturers for different types of solid, pneumatic and cushion tire, and also by wheel manufacturers on the subject of reducing operating cost. We must feel sorry for the operator who paid great attention to the various claims made for pneumatic tires in the last few years for trucks of 3½-ton capacity and over, although I believe that certain types of pneumatic tire can be employed to advantage for lighter equipment. Why should we consider tire cost only, in view of the differences in repair and mechanical-upkeep cost on various types of tire which indicate that a large saving can be made. Table 5 is illustrative of this point.

Based on the difference in cost stated in Table 5 and assuming that 100 trucks ran 25 days per month, 12 months per year and 35 miles per day, we have a total of 1,050,000 miles per year. The difference in the mechanical-repair cost in this case represents a saving of $32,970 per year, while the difference in the total mechanical cost represents a saving of $46,200 per year. These figures indicate why we should keep tire records and the reason for my contention that we should not arbitrarily consider one type of tire as good as another and figure tires on cost alone.

In connection with standardized fleet-operation cost, I suggest the necessity for a standard form of distribution of chassis-overhaul cost. The figures presented by Mr. La Schum are interesting, but it is difficult to make comparisons from the information submitted because the factors on which the costs are based are not stated clearly. Do his cost figures represent the cost of overhauling a 2, 3 or 5-ton vehicle, or do they represent an average of a number of vehicles? What mileage had the vehicle traveled before the overhauling was done? How old are the vehicles? How long did the vehicle operate after overhauling as compared with when it was new? These are questions that come to my mind in connection with the cost. My experience on one item in this classification leads me to believe that a 5-ton motor-truck, after traveling from 18,000 to 24,000 miles, should be overhauled on a labor-hour classification on an

TABLE 5—DIFFERENCES IN MAINTENANCE COSTS

		First Type	Second Type	Differences
Duration of Operation, Months	.....	27	25	.....
Total Maintenenace Cost per Mile	Tires.....	$0.0273	$0.0167	$0.0106
	Repairs.....	0.1043	0.0729	0.0314
	Mechanical..	0.1759	0.1317	0.0442

average of not over 90 hr., while the total repair-cost, including all charges, should not exceed an average of $900.

Maintenance Difficulties

The analysis of troubles shown by Mr. La Schum is very interesting. I believe the results of analyzing any particular fleet of trucks would depend largely on the types of vehicle employed. The type of vehicle, the road conditions and the average age of the fleet will make a great difference in the percentages. Disregarding these particular points, however, I seldom see or hear of trouble on the following items: Spark-plugs, carbureters, gasoline lines clogged or leaky, governors, propeller-shafts, universal-joints, radius-rods, drive chains, wheels, brakes out of order, gasoline and oil shortage or tires. Systematic inspection, strict attention to the education of the operator and methodical operation have eliminated the majority of the above complaints. Using a strainer funnel of the proper design, or a proper filter in the pump or in the nozzle of the hose from the pump, usually will prevent gasoline lines from becoming clogged or leaky. Instructing the driver in the use of spare chain links will take care of broken chains. Governors usually give trouble by failing to govern, but we never hear of an operator who is dissatisfied from this standpoint. Operators should be trained to detect spark-plug trouble; I seldom have trouble of this nature. My spark-plug consumption has been approximately one spark-plug per cylinder per year and the individual design of the oiling-system, and the skill of the operator play an important part in this. One type of car that we operate continuously has trouble with the forward cylinder. It is difficult to detect when this trouble is about to appear and, because of this difficulty, the spark-plug consumption on this particular design of car tends to make the fuel consumption higher than it would be ordinarily.

One of the greatest items of expense to the average operator is due to having vehicles out of service. Particular stress should be laid on this point and accurate records kept of the average time a vehicle is in operation. I believe that any cost system that fails to recognize this important fundamental will not give accurate results. Based on my experience, I believe that a fleet operating over an extended area and not coming into a central garage each night, a fleet that is left largely in the hands of an individual operator, except for a bi-weekly inspection, can be operated efficiently with a reserve equipment of approximately one truck for each sixteen trucks in service. If an operator has his vehicles under efficient central control and has a systematic method of upkeep, this reserve equipment could be reduced still further.

Future Practice

Although the vehicles in the industry are becoming more standardized from a standpoint of general construction, there are still many details and many practices that can be worked out to clear up some misconceptions that still exist. Continued tests of various grades of fuel should be made such as were made recently by the Society in cooperation with other organizations. Closer study should be made of the lubrication system by engineers, as there is a very wide variation in the lubricating-oil consumption per mile in different types of design. Magazine articles and instructions come to me having to do with the changing of lubricating oil at definite periods. The general figure mentioned by various companies is approximately from 500 to 600 miles. In checking-up these recommendations, it has been found that the amount of oil in the base of the crankcase exceeded, in some instances, that in others by at least 200 per cent. Provided that all other conditions are equal, the design having the 200 per cent excess of oil should not have the oil changed as often as is desirable for the model carrying the smaller amount. I hesitate to say so, but, based on my experience, I believe a certain magneto that was manufactured in 1914 was more efficient than the majority of the magnetos we have today.

Because of the type of work the vehicles that I supervise do, electric lights are employed for lighting purposes almost exclusively. There is need of paying more attention to the details of the wiring system, to the design of a generator suitable for charging a battery of the nickel-iron type; to a suitable mounting and a mechanical clutch that will permit the disconnecting of the generator when it is not operating. The electric generator should be built so that it will withstand the vibration of motor-truck operation.

There is need for an extensive study of semi-pneumatic or cushion-type tires, from the standpoint of carrying capacity. A suitable definition should be given as to what a semi-cushion type of tire is; and a definite and logical table as to the relative amount of damage done to the highways should be worked-out, so that State officials would differentiate, from a licensing standpoint, between the solid, the pneumatic and the semi-pneumatic tires. Then operators would pay in direct proportion to the damage caused by various types of equipment on varying models of tire.

The influence of the Society should be put behind the elimination of engine or unit numbers of any character, so that the fleet operator who desires to make repairs by the unit system can do so without running foul of the present laws. Invariably, business or passenger cars are licensed according to the engine number and trucks according to the chassis number but, in registering either type, the engine is always considered. Why will not the chassis number alone satisfy both the builder and the State authorities? The present system of making it a felony to change an engine number certainly complicates the prompt overhaul of trucks or vehicles on a unit-system basis.

F. C. Horner:—Several points in Mr. La Schum's paper are of great importance to successful motor-truck operation and should be considered thoroughly at every opportunity. First, what is the best way in which to decide what vehicles to standardize on? Mr. La Schum's experience is a splendid example of the problem of changing over a large number of miscellaneous motor-trucks, more or less nondescript in character, make, model and size, into a scientifically coordinated fleet in which standardization has been the main objective. The explanation of the conditions he describes was the merger of all the express companies into one, but little thought apparently had been given previously to standardization by the individual companies. It appears that, either through ignorance or from the unsatisfactory results with equipment already in use, it had been the policy of their transportation managers to experiment with every new make and size of vehicle brought out that seemed to be better than the older ones. The advantage Mr. La Schum had over his predecessors was that, through records of performance, he could eliminate the "duds" and gradually boil down his fleet to standards of make, model and size that would furnish the best service for the longest time and with the least interruption. But what plan should the beginner in fleet operation follow

in this matter of selection and standardization? I think it would be helpful to have the light of all the experience available thrown on this important question. Although not feeling competent to do so myself, I should like to have some of our technical men discuss Mr. La Schum's statement regarding "manufactured and assembled" trucks. Is one better than the other, when all things have been considered, or do they just about balance?

Careful Inspection Necessary

Another point in Mr. La Schum's paper is the vital necessity for careful inspection methods in motor-truck operation. The most thorough system of inspection in large fleet operation that I know of is to be found in the London General Omnibus Co. Naturally, when human life is at stake, too much care cannot be taken to avoid accidents but, coupled with the necessity for economical operation, maintenance and repair, inspection becomes doubly necessary in this case. The "Driver's Report and Dock Overhaul Sheet" is the basis of their inspection system. It covers a 30-day period and all involuntary stops are registered on the report. In another column "Defects and Remarks on Running" are registered, and in the column, "Repaired By" the mechanic making the repairs signs his initials. The month's record then is on one sheet, which is attached to the vehicle when it goes into "Dock" at the end of every 30 days. The foreman and his staff, by consulting the dock sheet, are able to determine the weak spots at once, if there are any, and to remedy them. Of course, even if the dock sheet showed a clean performance for the vehicle, the dock overhaul would be no less thorough. In addition, the inspectors go over every vehicle at night and make any repairs or adjustments that are necessary. One thing will show the thought given to inspection by this company. The steering-arms and the connections are kept highly polished and are wiped off each night with a rag wet with kerosene so that the inspector shall not fail to detect a flaw or failure in the metal, if one be present. It is the "stitch-in-time" methods that "save nine," as Mr. La Schum so aptly puts it. Unfortunately, too few motor-truck operators realize this important truth.

It is of interest to note that almost every motor-truck operator I met abroad referred to the unsatisfactory mileage recorders of today, and most of the recorders I saw in England were of American design. Why we cannot turn out an accurate, durable recorder is a mystery to me for, without accurate mileage records, the operator cannot keep dependable costs.

Engineering Advice

Another important point is the courteous manner in which Mr. La Schum deals with the engineers who design motor trucks. It has always been my opinion that automotive engineers could do more toward helping themselves as well as the truck operators if they knew more about the problems with which the operator is confronted. If the engineers of the truck companies would consult the experienced operators of motor vehicles and would profit by the knowledge gained, it would be unnecessary for Mr. La Schum to remind the Society of the things that have been left undone by the engineers, for many of which there can be no legitimate excuse. Simplification is an important thing that could be accomplished better by the engineer through closer cooperation with the operator and by a thorough knowledge of his daily problem. What engineers must do is to simplify their vehicles, make them better, but easier to get at, to inspect, to repair and to maintain and as nearly fool-proof as it is humanly possible to make them. Take off the frills; in service, essentials are the only things that count. On the other hand, what the operators must learn is that the "stitch-in-time" plan pays, that an ounce of thorough inspection is always worth a full-weight pound of repairs. A motor truck is a piece of fine machinery requiring proper care and attention on account of the varied nature of the conditions under which it functions as a means of transportation. Knowledge is power. Neither the engineer nor the operator can expect to reach a state of perfection without a thorough knowledge of the other's problems and close cooperation is the most effective way to obtain that knowledge.

Regarding the question of fuel consumption, a careful check of miles per gallon, by vehicle, by garage and by district, posted in such a way as to create a desire on the part of the garage superintendents and the drivers to lead the field, is one of the best ways to obtain high gasoline-mileage. Analyze your fuel from time to time to see that it is up to the mark. Chart your fuel-consumption records, study them and see where the leaks are, and, without adding frills to your organization, you will get good results.

On the subject of lubrication, it seems to me that it would be an easy matter for the truck builder to determine exactly what oil is best for the engine on his truck and to send out a formula of this oil with each vehicle. This formula would simply be a chemical analysis, so to speak, of what had been found to be the best oil for a particular engine. No mention of any brand or make of oil is necessary; the operator can check-up on his oil by chemical analysis from time to time, and the oil-supply company must toe the mark or lose the business. I tried this some years ago with good results.

E. E. La Schum:—Mr. Winchester said that he could not reconcile the gasoline costs. I think that this may have come about by the fact that I left out interest, depreciation, drivers' and helpers' wages and the like. I think the reason that gasoline cost runs high in operating express trucks is that these vehicles operate almost entirely in congested traffic, running on change gears, backing-up to and pulling away from shippers' platforms and doing a large amount of engine-running while accomplishing no mileage. One of the most difficult things to determine and to prove is whether a cushion wheel, a pneumatic tire or a semi-pneumatic tire is better than a solid tire, because express trucks run exclusively in cities. I can say without hesitancy that road conditions generally are smooth. We do not get vibration, but the fellow who runs between New York City and Philadelphia at a high rate of speed cannot get speed and not get vibration.

W. P. Kennedy:—In Table 1, why are the body and the chassis-repair costs 40 per cent when the gasoline cost is 30 per cent?

Mr. La Schum:—The truck is doing a great amount of running that brings about wear and tear. The body repairs are more apt to be necessary in congested territories, on account of collisions with wagon poles and with each other, when backing-up in close quarters at railroad depots. If it were not for this item of collisions, the body repairs would not amount to nearly so much.

Mr. Kennedy:—What is the 2-ton electric truck mileage without boosting?

Mr. La Schum:—A 2-ton electric truck in proper condition, with the loads varying as they do in the express service, usually will make about 30 to 35 miles per day on a single battery-charge.

MR. KENNEDY:—Have you made any observation as to variation in the tire-maintenance cost due to the effects of different road impacts?

MR. LA SCHUM:—No.

R. E. PLIMPTON:—Will Mr. La Schum tell us how he determines the period of overhaul on which the costs are based? Does it depend on mileage, or time, or on the judgment of the shop superintendent?

MR. LA SCHUM:—It depends on general judgment, but mileage has more to do with overhaul than anything else; approximately, it is done after each 18,000 miles of travel. The trucks to which the costs given apply were two 3½-ton trucks.

CHAIRMAN W. E. KEMP:—Mr. Horner asked you to define assembled and manufactured trucks more clearly.

MR. LA SCHUM:—I have said that the terms "manufactured" and "assembled" resolved themselves into one word. I cannot understand what the difference is, because all trucks are manufactured. Some builders manufacture one unit and some another. The point that strikes me forcefully is that, if you dismantle a so-called manufactured or a so-called assembled truck, and put it on one table, then dismantle another truck and put it on another table, a layman can pick out parts that are identical such as the wheels, radiators, frames and the like. After the layman has picked out the parts that are identical, if you call in a well posted man he will pick out more parts. An engineer will pick out parts, not identical in shape, but identical in function. After you have finished, it seems to me you will have a manufactured truck.

One thing that engineers could do, if it were consistent with good salesmanship, would be to indicate to the truck user that if a part is broken he does not need to throw up his hands just because there is not a service-station next door. There are many sources of supply that will enable him to obtain that broken part. Service-stations are necessary and truck builders and truck assemblers have done much at great expense to bring about the establishment of proper service-stations but, if I have a broken spring, I do not necessarily think of the truck builder, because I know of many places where I can procure the same kind of spring; if I have a broken magneto part, there are plenty of places in which to get the magneto repaired. Ball and roller bearings are interchangeable in a dozen different kinds of trucks. That is true to a considerable extent of engine and connecting-rod bearings, though not so universally. It is more true with assembled trucks than with so-called manufactured trucks. Still I do not see the difference between so-called manufactured and assembled trucks.

W. R. GRAHAM:—Has Mr. La Schum tried pneumatic tires on electric trucks?

MR. LA SCHUM:—We never use pneumatic tires on electric trucks because they are slow and, because of the continuous torque, it has never been considered necessary to try them.

MR. GRAHAM:—What plan should a beginner in fleet operation follow in the matter of selection and standardization?

MR. LA SCHUM:—It is a fact that all the old express companies, before they were merged into one, did considerable experimenting. No industry has done more to help the motor-truck industry, because the express companies spent their money freely and tried many types of truck, not through ignorance, but to assist in promoting the motor-truck industry. After the motor-truck builders had reached a position where they could stand on their own feet and fight for business, the reason the American Railway Express Co. inherited fifty-nine different makes of trucks and 131 different models was reciprocity. All the builders took advantage of the express company because they shipped everything by express. They continued to use the big stick of reciprocity until 1918, when the Government told the express company that it must get down to a standardized basis and give more economical performance.

The so-called assembled truck simplifies the maintenance problem. So many units are identical that, although the trucks are built by different companies, I can maintain them economically.

A MEMBER:—What does Mr. Winchester consider to be the relative value of grease versus oil for a gearbox or differential lubricant?

MR. WINCHESTER:—I can make no relative comparison because I only use and recommend oil for this purpose.

In connection with my comparison of gasoline consumption with that of Mr. La Schum, I stated that the gasoline cost was 33⅓ per cent of the total maintenance cost. I say that because, when an average man thinks of 33⅓ per cent of his costs being gasoline, he is apt to be critical of the oil company. In my consideration of this problem, as is shown in Table 4, I have included under fixed charges: Chauffeur and help, storage, insurance, license and depreciation, which constitute the 65.2 per cent of the total. Under variable charges, I have included gasoline, lubricating oil and grease, tires and road and miscellaneous expenses. When I get on the same basis that Mr. La Schum has considered and take into consideration gasoline, oil and grease, body and chassis repairs and garage expenditures, I find that my figures practically agree with those of Mr. La Schum; if anything, they are a little higher on the gasoline consumption.

Research Topics and Suggestions

THE Research Department plans to present under this heading each month a topic that is pertinent to the general field of automotive research, and is either of special interest to some group of the Society membership or related to some particularly urgent problem of the industry. Since the object of the department is to act as a clearing-house for research information, we shall be pleased to receive the comments of members regarding the topics so presented, and their suggestions as to what might be of interest in this connection.

INFORMATION SERVICE

THE importance to engineers of having readily available a fund of technical information pertinent to the solution of their problems can hardly be over-estimated. Such information forms a basis for the development of new ideas and makes possible the extended application of principles already established. Its usefulness is largely governed by its degree of availability.

There are those who prefer to launch a project or enter a new field of investigation entirely uninfluenced by that which has already been accomplished in identical or closely related fields of endeavor. But it is certain that a great majority of workers believe that thorough enlightenment is the first requirement; they desire at the outset to be fully informed as to past activities and through this information to be warned against the pitfalls that have retarded the progress of others.

In spite of the generally accepted validity of the above statements, however, each individual organization in the industry does not maintain and probably cannot reasonably be expected to maintain a complete or even very comprehensive repository for information. Although some have gone far in this direction by establishing extensive libraries and reference files, yet the vast majority still depend largely upon the efforts of the individual to glean his facts wherever he may, from sources that are often uncertain and scattered and, in some cases, inaccessible.

It was with a realization of these conditions and in consequence of a lively and increasing demand for technical information that the Society of Automotive Engineers' information service was established some years ago. It has since become an important part of the activities of the Research Department. From its inception, this branch has undergone a continued enlargement of its sphere of activities, as the demands made upon it have increased. It has always been the aim to render an important service to the members by making available, and readily so, any obtainable material on automotive and allied subjects that may be of particular interest.

Many requests for such information are answered on the day of their receipt; they may involve questions with which members of the research staff are familiar and that can be answered simply, or they may require references to books or other publications that are cataloged in the reference files. In this connection, it should be stated that, in addition to being entered in the reference files, all important articles in publications received each month are carefully perused and those of sufficient value are either mentioned or reviewed in detail in the Notes and Reviews column of THE JOURNAL. References to articles on various subjects are sent to members who have expressed a desire for them.

Certain inquiries demand more extensive investigation than the type mentioned above, before they can be properly and adequately answered. For such work there is accessible the Society's library that contains a large number of books on automotive and related subjects and also a file of over 60 periodicals, both domestic and foreign. The use of this library is directly available to members. In addition, access may be had to other libraries, notably that of the Engineering Societies with its very extensive collection of publications. Furthermore, the Department has a file containing a great amount of technical information obtained by the research staff through personal consultation with various investigators who have seen fit to release the material for distribution. This file includes, besides printed documents from governmental and educational organizations, a large number of informal reports on various subjects.

Inquiries are frequently received that relate to subjects not covered by publications or documents at hand but that are of sufficient general importance to the membership to warrant making a special investigation. This involves correspondence as well as interviews by a member of the research staff. To illustrate: a firm recently addressed the Department, stating that trouble was being experienced in finishing the surfaces of piston-ring grooves to a high degree of smoothness and accuracy. It was explained that the ordinary forming tool leaves in the metal small ridges, whose initial wear in service being very rapid, causes looseness of the rings and corresponding leakage past them.

A number of letters, stating the above case, were sent to leading manufacturers with a request that any person who had available for distribution material of interest on this subject send it to the Department at his convenience. The returns from this inquiry and reports of personal interviews on the subject have been forwarded immediately to the inquirer. Furthermore, it is expected that sufficient material of general interest will be received to warrant its presentation in these columns at an early date. Obviously the success of an investigation of this type is very largely dependent upon a spirit of cooperation and helpfulness among engineers. This has been found to exist in a marked degree.

It is the policy of the Department to maintain, so far as possible, a direct contact with persons engaged in research, with a view to assisting them by furnishing needed information. This is often accomplished by personal visits by members of the staff.

It is hoped that increasing numbers of the membership will take advantage of the benefits of the information service and that the present hearty support of all will be continued.

Tractor and Plow Usage in the First Breaking of Peat Land

By J. Lyman Larson[1]

Minneapolis Section Paper *Illustrated with* Photographs and Charts

A TRACT comprising several acres of "tamarack" swamp, drained with tile the previous fall and cleared of stumpage the following spring, was utilized to obtain accurate information regarding the tractor and the plowing equipment required for the heavy operations of first breaking of the peat soil, which was from 5 to 7 ft. deep. The paper describes the equipment used, gives details of the procedure and presents the data that were obtained.

Three different types of tractor and two types of plow, the latter having either a marsh or a breaker bottom and equipped both with and without a furrow pusher were used. Specifications of the tractors and the plows are given and commented upon, graphs and tabulations of the results being presented also. Power requirements on timbered peat and on grass marsh are compared and the efficiency of the plowing equipment is discussed.

INFORMATION regarding the equipment necessary for peat-land farming is both meager and conflicting. This is true more particularly with respect to the preliminary operation of breaking and preparing the first seed bed. A tract of peat land under the control of the Minnesota Agricultural Experiment Station was being put under cultivation for drainage experimental purposes, in the spring of 1922, and the opportunity was therefore utilized to obtain accurate information regarding the equipment required for the heavier operations and the best methods for applying that equipment.

Fig. 1—The Character of the Original Timber and the Quantity of Stumps Removed

The tract selected comprised several acres of "tamarack swamp" drained with tile the previous fall. The peat on the tract was from 5 to 7 ft. deep and had carried a very heavy growth of trees, principally tamaracks, birch, elm and maple. The clearing was done between April 15 and June 15 and it was estimated that 50 cords of stumps per acre were removed. Fig. 1 shows the character of the original timber and the stumps. The holes left by the removal of the stumps and the soft spots left by the back filling of the drains were very considerable factors in all the cultural operations, more particularly in the first plowing. It was considered that the job was a fairly difficult one, but one that was thoroughly representative of the actual conditions which would have

Fig. 2—How a Tractor Became Bogged at the Beginning of the Plowing Because of Insufficient Support

to be met by the farmer in breaking-up timbered peat land. Fig. 2 shows a tractor bogged at the beginning of the work and Fig. 3 a horse-drawn implement and team standing on this identical spot after it had been plowed and rolled. During the progress of the experiment, the water level below the surface of the peat was maintained at about 3 ft. at the lower end and 4 ft. at the upper end. The soil was fairly firm at the upper end of the tract, and had a considerable amount of sod; at the lower end, the peat was very soft and much torn and pitted by the removal of the stumps.

G. R. B. Elliott, who designed and was in charge of the general project, utilized a ditch that already was on the property and controlled the water at six different levels, with a 1-ft. difference between levels, the idea being to get accurate knowledge of the effect that the water level has on the crop. He obtained variations of from 400 to 1000 per cent due to water level. Weir boxes were installed in the ditch at various places; flash-board boxes were used inside the weir boxes to hold the water up to the desired levels.

Plowing Depth

A very important consideration in the first breaking of any land is the getting rid of trash and soft litter. On peat land it is also especially important to plow sufficiently deep to get under the deep underground rootstalks of marsh vegetation. A very large proportion of swamp plants are perennials that store food for the next spring's growth in thickened roots. For example,

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heavy root stalks of nettles may frequently be found 1 ft. below the surface, and cat-tails may have theirs 15 to 18 in. deep. The greater number of heavy root-stalks, except those of grass and small sedge, are to be found in the second 6 in. below the surface; therefore, 10 to 12 in. is about a minimum depth for first plowing under

FIG. 3—A HORSE-DRAWN IMPLEMENT STANDING ON THE SAME SPOT WHERE THE TRACTOR IS SHOWN BOGGED IN FIG. 2

average conditions on peat swamps other than grass marsh. Hence, plows were chosen that would cut effectively to that depth; namely, plows with 16 to 20-in. shares. These required comparatively heavy tractor equipment. A total of five types of plow and three types of tractor were used.

One of the facts that caused the decision in favor of deep plowing was that of the second-year-crop condition of soil. Clover planted in the fall on an 8-in. plowing after a flax crop had attained a much ranker growth by the latter part of the following May on the soil that had been broken 12 in. deep the first year than on soil broken only 6 in. deep.

TYPE OF TRACTOR

Three different types of round-wheel tractor were available. These varied in design, weight, width, operating speed and special equipment.

Fig. 4 shows a tractor properly equipped for marsh plowing. It has deep skid-rings on the front wheels, wide extensions, widely spaced angle-lugs and spade teeth on the driving wheels. This does not mean that

TABLE 1—SUMMARY OF TRACTOR DETAILS[2]

	Tractor		
	A Two-Plows	*B* Three-Plows	*C* Three-Plows
Power, hp	18 (Belt)	12 to 25	12 to 20
Plowing Speed, m.p.h. Low	1.48	2.25	2.10
Plowing Speed, m.p.h. High	2.73	3.00	2.75
Overall Width as Equipped, in.	61¾	84	79
Wheel Dimensions, Driving, Standard, in.	12x42	12x52	12x54
Wheel Dimensions, Driving, As Equipped, in.	12x42	18x52	24x54
Wheel Dimensions, Front, Standard, in.	5x28	6x28	6x36
Weight, lb. Complete	2,920	6,100	5,120
Weight, lb. On Front Wheels	1,063	1,575	1,583
Weight, lb. Per Inch of Front-Wheel Width	106	131	132
Weight, lb. On Driving Wheels	1,857	4,525	3,352
Weight, lb. Per Inch of Driving-Wheel Width, as Equipped	77.40	125.50	69.83
Design for Wheels To Be on	Furrow	Furrow	Sod
Drawbar Offset, in.	11	10	10
Front Skid-Ring Depth, as Equipped, in.	1	1½	3
Angle-Lug Spacing, in.	7	10	20
Depth of Angle Lugs, in.	3½	3	3
Depth of Regular Involute Spade Lugs, in.	.....	.....	4
Number of Teeth Between Lugs	.....	.....	16

[2]Used a 3000-lb. Shaffer & Budenburg Co. dynamometer.

TABLE 2—SUMMARY OF PLOW DETAILS

Details	Types of Plow				
	XB, 20-In.	*YM*, 20-In. With Furrow Pusher	*YM*, 20-In. Without Furrow Pusher	*XB*, 16-In.	*YB*, 16-In.
Width of Cut, in.	20	22	22	16	18
Weight, lb.	1,010	905	805	927	805
Wheel Mounting	Two	Two	Two	Two	Two
Wheel Width, in.	4	6	6	4	6
Overall Length, ft.-in.	13–10	13–10	13–10	13–10	13–10
Throat Clearance, in.	24	23½	23½	24	23½
Drawbar Adjustment	Triangular	Adjustable	Tongue	Triangular	Adjustable Tongue
Length of Plowshare, Point to Heel, in.	33½	31	31	28½	25¾
Landside Length, in.	69	60½	60½	69	52
Landside Height, in.	6	6	6	6	3
Vertical Tangent Height, in.	16	12	12	11	11½
End of Mold board to Plow Point, in.	58	54½	54½	55½	50
End of Mold board to Plow Heel, in.	32	27½	27½	28	28

any particular tractor is better for this work; it means only that it has the proper equipment. Some of the tractors failed solely because they lacked the proper equipment for such work.

As the type and equipment of the tractors had as much to do with the quality of work done as did the character of plow drawn, the specifications of the tractors and their equipment are given in Table 1. For convenience, the tractors are designated *A*, *B* and *C*. The plows are designated *X* and *Y*, followed by *M* or *B* to indicate whether a marsh or a breaker bottom was being used; plow details are given in Table 2.

Tractor *A* was available from the beginning, but it was felt beforehand that it would not pull the plows; so, no attempt was made to use it on this part of the work. The horsepower indicated by the dynamometer readings made with the heavier tractors shows this opinion to have been correct. Tractor *A*, however, was used to help Tractor *B* by hitching it on in front as described later. The mode of attaching the dynamometer is shown in Fig. 5.

Tractor *B* did not come on the ground properly equipped in all respects, and its use cannot be considered a success. It was as powerful as Tractor *C* and, with similar wheel equipment, undoubtedly should have done the work just as well. Its rear wheels carry a much heavier load per square inch of bearing surface than do either Tractors *A* or *C* and, while carrying the same load

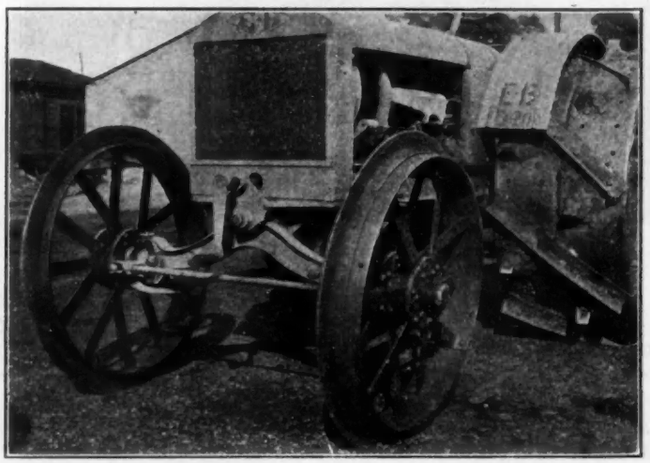

FIG. 4—A TRACTOR PROPERLY EQUIPPED FOR MARSH PLOWING

FIG. 5—METHOD OF ATTACHING THE DYNAMOMETER TO THE TRACTOR

per width of face, its front wheels were much smaller in diameter. The depth of skid rings on the front wheels was practically negligible. The material traveled over was very soft and slippery; consequently, the driver had little control over the direction of his machine and, when the front end plunged into the soft back-fill of a ditch or into the hole left by the removal of a stump, the tractor almost invariably mired and had to be pulled out with the help of Tractor *A*. Finally, Tractor *A* was kept hitched on ahead by a chain about 15 ft. long.

Tractor *B* was first hitched to a 20-in. *XB* plow and, at the instance of the factory operator, it was tried by running with the two right-hand wheels in the furrow. One length of the field was completed after much difficulty and miring, but the plow was not actually plowing at any time. After this, the drawbar of the tractor was pushed as far as possible to the right, while the hitch of the plow was thrown to the left. This permitted the tractor to travel on the sod or unplowed ground, but the best furrow that could be cut was about 6 in. in depth and there was a heavy sidedraft that tended to pull the tractor into the furrow. As there was nothing to prevent the front wheels from skidding, this slewing of the tractor was serious. It is probable that by using a chain hitch, as was done later, the side-draft could have been eliminated partially. Figs. 6 and 7 show, respectively, the drawbar adjustments of the *XB* and the *YM* plows.

The greatest trouble was experienced in turning at the ends of the field. The front end of the tractor had no grip on the soft ground and, when a wheel fell into a hole, the force of the tractor would drive it still deeper into the far side. In trying to back the tractor, much the same thing happened and the rear wheels frequently mired by first cutting-up the sod with the closely set angle-lugs. Tractor *A* was then hitched on ahead to help in the steering and to prevent Tractor *B* from getting mired. This arrangement worked satisfactorily and the plow cut to a depth of 8 to 12 in. As these were not standard operating methods and as the proper wheel equipment could not be obtained readily to hold-up Tractor *B*, it was decided to discontinue the work in the hope that a tractor could be found with approximately as much power but with less weight per inch of width of the wheels, one having the equipment necessary to make it operate successfully and with its drawbar designed so that it would permit the tractor to run on the sod and without undue side-draft. We were able to get a tractor that seemed to answer the requirements, the makers of which were willing to cooperate in the work under way. This is Tractor *C*. Fig. 8 shows the offset of the *XB* plow; that of the *YM* plow, equipped with a sod pusher, is shown in Fig. 9.

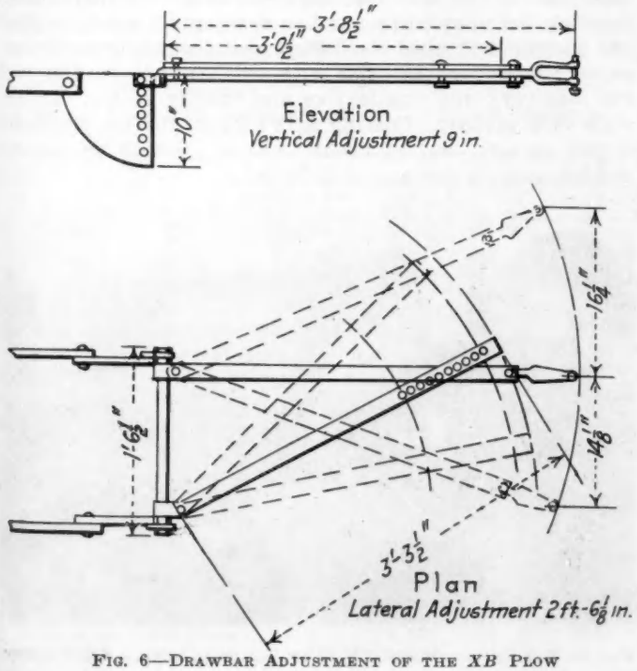

FIG. 6—DRAWBAR ADJUSTMENT OF THE *XB* PLOW

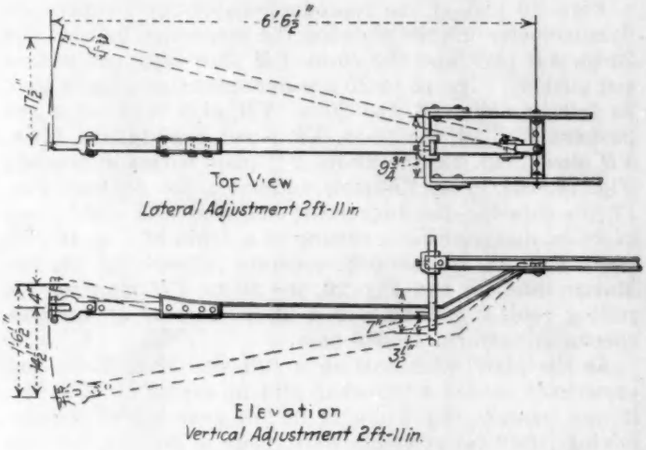

FIG. 7—DRAWBAR ADJUSTMENT OF THE *YM* PLOW

Profiting by previous experience, we asked for 12-in. extensions on the rear wheels, 4-in. extensions on the front wheels, skid rings on the front wheels 3-in. high, angle-iron lugs that would run clear across the face of the rear-wheel rims, placed with wide spacing; and 4-in. spade-lugs in pairs between the angle lugs. With the tractor used and owing to the comparatively light weight on the front wheels, it was found that it was not necessary to use the 4-in. extensions. Instead of hitching the plow directly to the drawbar of the tractor, as was the case with Tractor *B*, the drawbar was moved to the right almost as far as it would go and the hitch of the plow only slightly to the left. Then a 3-ft. length of chain was used to couple the plow to the tractor. This practically eliminated the side-draft on the plow. The hitch was made so that it could be dropped quickly and the

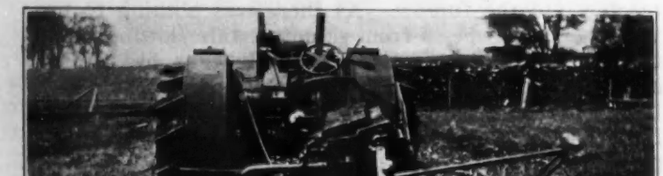

FIG. 8—THE OFFSET OF THE *XB* PLOW

chain extended to 15 ft. In case of miring, the tractor could run ahead to a better footing, pull the plow through the soft spot and then back up to the short hitch. This method proved satisfactory and the work was finished in this way.

Figs. 10 and 11 are reproductions of the comparative dynamometer graphs showing the respective pulls of the 20-in. *XB* plow and the 20-in. *YM* plow equipped with a sod pusher. Figs 12 to 20 are dynamometer graphs also, as follows: Fig. 12, the 20-in. *YM* plow without a sod pusher; Fig. 13, the 16-in. *XB* plow; Fig. 14, the 16-in. *YB* plow; Fig. 15, the 20-in. *YM* plow with sod pusher; Fig. 16, the 16-in. *YB* plow without a sod pusher; Fig. 17, the cutaway-disc implement equipped with eight discs of 20-in. diameter each, cutting to a depth of 7 in. in soft peat; Fig. 18, the 3500-lb. concrete roller; Fig. 19, the Martin ditcher; and Fig. 20, the 20-in. *YM* plow with a rolling coulter cutting a 9 x 22-in. furrow at variable speeds in uniform sodded peat.

As the plow when cutting a full size 12 x 20-in. sod sometimes caused a drawbar pull in excess of 2000 lb., it was thought that running in low gear might prevent miring. Several attempts were made to do this, but the interesting fact developed that the tractor mired far worse when running in low gear than in high. It would seem that the momentum of the higher speed or the short time the machinery rested on the soft footing permitted it to carry across the soft spots. The widely spaced angle-lugs and long spade-lugs gave a much better grip on the sod without cutting it up, than did the angle-lugs that were spaced more closely. Tractor *C* did nearly all the later work. It was found that after the ground was harrowed and rolled, Tractor *A* would handle the drag harrow, the log drag, the Martin ditcher or any of the lighter machinery; but it would not pull the heavy roller or the cutaway-disc implement loaded to cut to a sufficient depth.

PLOWING DETAILS

In connection with the plow details presented in Table 2, the first trial was made with a 20-in. *XB* standard breaker-bottom plow equipped with a standing coulter. This was the same plow that was used with Tractor *B* in the early plowing, using a rigid hitch. In the tests, however, a chain hitch was used and only one tractor, so as to eliminate all variables such as the speed and the difference in the hitch.

As stated in Table 3, which shows the power consumption of the plows while working in timbered peat, when cutting an 8 x 20-in. furrow, the 20-in. *XB* plow developed a drawbar pull of 1350 lb. in the tests. At a 10-in. depth, the drawbar pull increased to 1500 lb. for a 20-in. furrow and 1650 lb. for a 22-in. furrow. At a 12-in. depth, the drawbar pull was 1800 lb. for a 22-in. furrow.

The 16-in. *XB* plow had a standard brush-breaker bottom. It was tested at 8 in. of depth in firm peat and developed a 1050-lb. drawbar pull; at 10 in. of depth it developed a 1350-lb. drawbar pull. In soft peat, with a cut 10 in. deep, the drawbar pull required was 1200 lb. The furrow was 16 in. wide throughout.

The next plow used was a 20-in. *YM* marsh plow equipped with a furrow or sod pusher and a 24-in. rolling coulter. The center of the coulter was set 3 in. back from the point of the plowshare, and the cutting edge lacked 4 in. of penetrating as deeply. There was no adjustment that permitted a deeper or more forward setting. In use, soft roots were passed under the coulter in great quantity and clogged the throat after only a few feet of advance, as shown in Fig. 21. The rolling coulter was then replaced by a standing coulter. All later work on the timbered peat was done with this type of coulter. It is doubtful if the rolling coulter would have been successful in this type of ground, even had adjustments been possible. The plowing efficiency and power consumption of the 20-in. marsh plow at various speeds are shown in Fig. 22.

The sod or furrow pusher shown on the plowshare of the Type *M* plow in Fig. 23 is not, as may be supposed at first, merely a moldboard extension. It is a curved steel piece attached inside of the moldboard near the point of the plow and extending backward below and behind the moldboard on a level with the bottom of the furrow. It is cross-braced at the back against an enlarged and strengthened landside. It was developed largely in cooperation with the Agricultural Engineering Department of the University of Wisconsin, solely as an aid in marsh plowing. Its purpose is to push the freshly turned furrow still farther to the right, so that the next succeeding furrow will fall absolutely flat and then be pushed tight up against its fellow, completely covering the grass and other litter as illustrated by the finished plowing shown in Fig. 24.

The first trial with the plow was made with a rigid hitch, but this method had to be abandoned; when the driving wheels of the tractor dropped into a depression, the beam of the plow was carried down with them, thus burying the plow bottom and increasing the draft so that the tractor destroyed its footing and the whole outfit became mired. In the trial with this equipment, the sod cut was very uniform in size and the lay of the furrow slice very perfect. Cutting a 12 x 24-in. furrow required a pull in soft peat of about 1650 lb., and in the dense sodded peat an average of 1950 lb.

FIG. 9—THE OFFSET OF THE *YM* PLOW EQUIPPED WITH A SOD PUSHER

The same plow was then tried without the furrow pusher. The pull was an average of 1650 lb. both on the soft and on the firm peat. Comparing the pull of the plow with and without the furrow pusher shows no increase in the soft peat, and a 300-lb. increase where a solid furrow-slice has to be pushed to one side. The increase in the power required was more than compensated for by the very greatly improved quality of the work done. During this trial, the drawbar of the plow was also shifted to give a direct draft. The pull required dropped 150 lb., but the tractor could only be held on the sod with difficulty.

In all the trials with the 20-in. *YM* plow, it was noted that soil packed in the center of the curve of the moldboard, particularly toward the front. Apparently this packing of the soil was due to a sharper curvature of the moldboard of this plow than of the others, for the other plows did not clog in the same way and no change in the hitch or the adjustment of this plow made it scour through the center except where the sod was more than usually firm and dense. Again referring to Table 3, the 16-in. *YB* plow was a standard brush-breaker bottom applied to the same frame and wheels as those of the 20-in. *YM* plow. At 18 in. of furrow width and 12 in. of depth, the drawbar pull was an average of 1250 lb., at 20 in. of width and a similar depth, the pull averaged 1350 lb.

Other Machinery

A very valuable implement for peat cultivation is the heavy roller for packing the seed bed. The one used was home made and is illustrated in Fig. 25; the roller was of steel filled with concrete and had a steel frame with a diameter of 3 ft., a length of 4 ft. and a total weight of 3500 lb. When moving the pull developed was an average of 950 lb.; however, when the roller stopped, it settled into the peat and, in each case, a pull of 2300 lb. was required to start it.

The cutaway-disc implement proved itself very valuable in breaking-up the seed bed. It was of the standard 5-ft. width and had eight 20-in. discs. It had to be loaded with about 200 lb. to make it cut 7 in. into the ground. The pull developed was 1050 lb., and a 1500-lb. pull was required to start the load. This machine, even when loaded to make it cut deeply, would not do good work with one passage over sod land. Wherever the

TABLE 3—POWER CONSUMPTION OF PLOWS IN TIMBERED PEAT

Type of Plow	Type of Soil	Size of Furrow, In.	Cross-Section, Sq. In.	Drawbar Pull, Lb.	Drawbar Pull, Lb. per Sq. In.	Speed, Ft. per Min.	Drawbar Hp.
XB, 20-in.	.....	20x8	160	1,350	8.4	242	9.92
	.....	20x10	200	1,500	7.5	242	11.03
	.....	22x10	220	1,800	8.2	242	13.23
YM, 20-in. with Sod Pusher	In Soft Peat	24x12	288	1,650	5.7	242	12.13
	In Sod	24x12	288	1,950	6.8	242	14.34
YM, 20-in., Without Sod Pusher	In Sod and Soft Peat..	24x12	288	1,650	5.7	242	12.13
	On Direct Pull	24x12	288	1,500	5.2	242	11.03
YB, 16-in.	.....	18x12	216	1,250	5.8	242	9.19
	.....	20x12	240	1,350	5.6	242	9.92
XB, 16-in.	In Firm Peat	16x8	128	1,050	8.2	242	7.72
	In Firm Peat	16x10	160	1,350	8.4	242	9.92
	In Soft Peat.	16x10	160	1,200	7.5	242	8.82
Cutaway Disc.....		60x7	420	1,050	2.5	242	7.72
Roller.....		.....	.....	950	.....	242	6.98
Maximum Pull on Plows..		.....	.....	2,400	.....	242	17.64

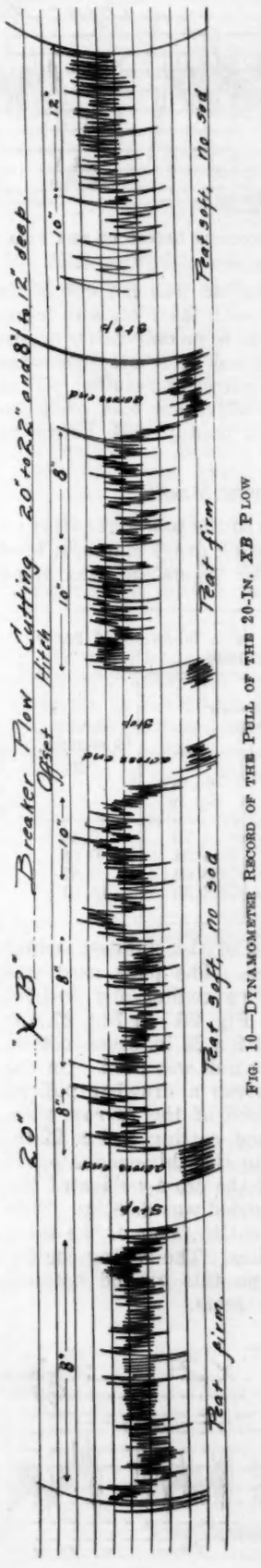

Fig. 10—Dynamometer Record of the Pull of the 20-In. XB Plow

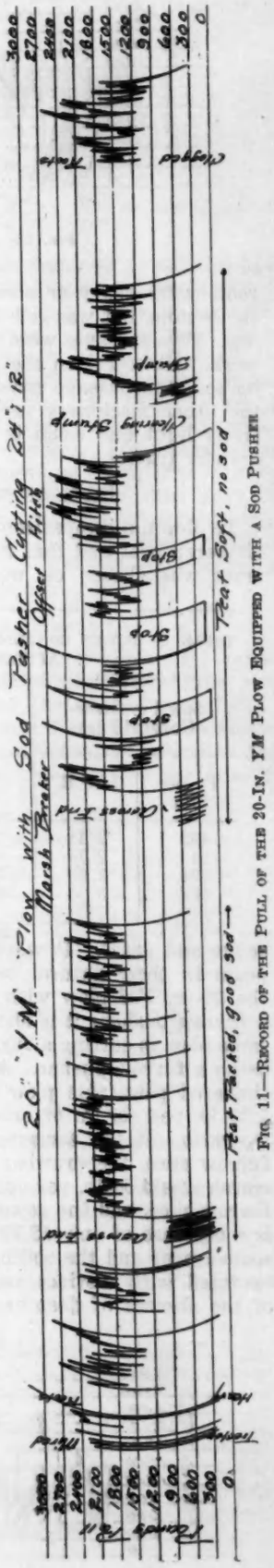

Fig. 11—Record of the Pull of the 20-In. YM Plow Equipped with a Sod Pusher

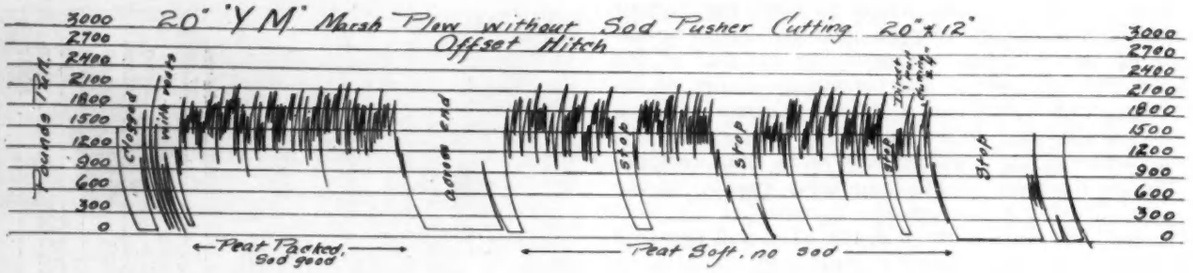

FIG. 12—DYNAMOMETER RECORD OF THE PULL OF THE 20-IN. *YM* PLOW WITHOUT A SOD PUSHER

roots were heavy or strong, the sod was not cut off in the bottom, but was rolled over with the roots still hanging. Five passages were required to do only fairly decent work. However, on the turned sod or where there was no sod, one passage gave very excellent results. All of the other machinery used in cultivation was comparatively light and could be drawn readily with horses or a light tractor.

TESTS ON THE OPEN MARSH

To obtain a comparison between the power required on timbered peat and that required on grass marsh, a level tract was chosen on which the vegetation was small sedge and grass. It was typical of the thinnest sodded peats in their natural condition. The plow used was the 20-in. *YM* plow with a rolling coulter and without a furrow pusher; it is shown in Fig. 26. In Fig. 27, the same plow is cutting a furrow of 9 x 22 in. cross-section, using a furrow pusher. A direct pull was used. On the timbered peat, this plow had given a drawbar pull of 5.20 lb. per sq. in. of cross-section of the furrow slice. Operating at the same speed and cutting a 9 x 22-in, furrow slice, the drawbar pull in the thin-sodded open-marsh was 4.54 lb. per sq. in. of the cross-section of the furrow slice, and the power expended was 6.68 hp. This is a decrease of only 12.70 per cent in favor of the thin-sodded peat and the rolling coulter. The rolling coulter operated with distinct success on this ground because of the absence of deep or heavy roots.

TABLE 4—POWER CONSUMPTION OF A 20-IN., *YM* PLOW AT VARYING SPEEDS

Speed in Grass-Covered Peat		Drawbar Pull		Power Consumed, Hp.
Ft. per Min.	M.P.H.	Total, Lb.	Lb. per Sq. In.	
192	2.19	950	4.79	5.55
245	2.78	900	4.54	6.68
324	3.68	1,000	5.05	9.82
522	5.93	1,050	5.30	16.61

The tractor was then operated at varying speeds to test-out its plowing efficiency. In all, four speeds were used of approximately 2.20, 2.75, 3.75 and 6.00 m.p.h. It was found that the efficiency of the plow decreased with the slower speed and also with the higher, although, in the latter case, the greater drawbar pull must be offset against the greater acreage plowed per day. Table 4 shows the power consumption of the 20-in. *YM* plow at ranging speeds when cutting a 9 x 22-in. cross-section in grass-covered peat. Fig. 28 illustrates the completed work.

In conclusion, several important factors were brought out by the trials. What can be called the fundamental consideration upon which all the others are built-up is the better quality of work done by the heavier plows. This was confirmed later by the lessened weed growth on the deep plowing. During the summer, nettles were plentiful on the shallow plowing but not on the rest of the field. The manner in which the land was cultivated later did not permit tests for crop yield on the various depths. It is not assumed that deep plowing on new peat will help the crop yield the first year; the reverse is probably true. There is no question, however, that it gets rid of the old growth and puts the land in better condition forever after. Throughout the whole test, the larger plows did better work than the small ones and gave a more complete turning of the furrow.

The rolling coulter as it was equipped was certainly not a success on the timbered peat, and it does not appear how it can be made a success on peat containing roots too heavy to cut or so soft that they will bend easily. To be got rid of, these roots must actually be torn out of the ground. On open marsh sodded with shallow rooting, small sedges and blue-joint grass, the 24-in. rolling coulter was a success, and possibly a smaller plow would also be satisfactory. In marsh containing reeds and the larger sedges, it probably would be necessary to have the coulter sufficiently large to penetrate below and forward of the point of the plow, in this manner either cutting off fibrous roots or carrying them into the bottom of the furrow.

The furrow pusher was a distinct success. It did not add as much to the draft of the plow as was expected; it pushed the newly turned furrow out of the way and

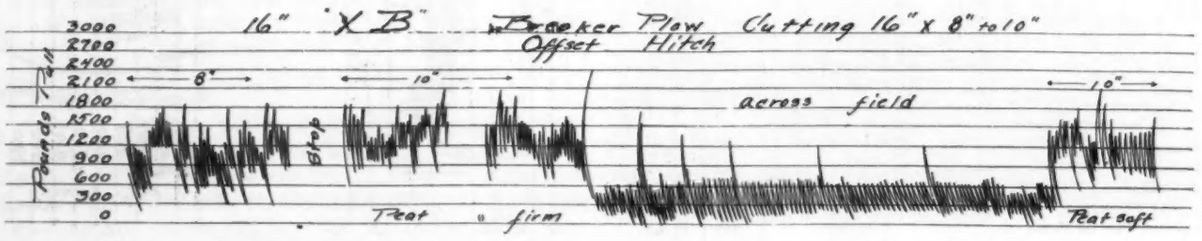

FIG. 13—THE RECORD MADE BY THE DYNAMOMETER OF THE PULL OF THE 16-IN. *XB* PLOW

permitted the more complete turning of the sod and the elimination of top litter.

Power Required

An analysis of the power consumption developed several interesting facts. On the larger plows, the drawbar pull was almost constantly near or above 10 hp., with a reserve up to 17 hp. that was drawn upon at times. For the smaller plows, the power consumption was above 8 hp., with a reserve up to 15 hp. that was called upon at times. From this it would appear that, for marsh breaking, a tractor is necessary that is capable of developing not less than 12 hp. as a working drawbar power, with a reserve above that. This power and a power-reserve is not necessary for later plowing.

Special Wheel Equipment

Ample support to the tractor proved to be just as important as power. Tractor *B* failed rather signally, not

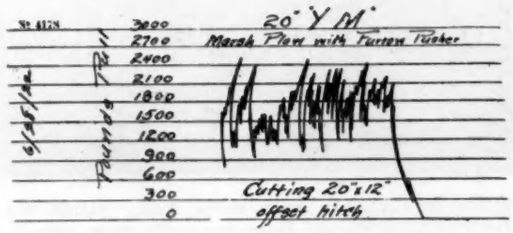

Fig. 15—Another Record of the Pull of the 20-In. *YM* Plow Equipped with a Sod Pusher

from any inherent defect in the design of the tractor itself or from a lack of power, but because the special wheel equipment that would enable it to do its work was not available. It was like a strong swimmer thrown into the water and told to swim with stones tied to his head and to his feet. It was capable of developing only about a 1600-lb. drawbar pull without miring, though it was able to do much more on hard ground. It not only mired its front wheels but, when the plow stuck, the closely spaced angle-irons on the driving wheels cut the sod, which was then thrown backward, leaving the tractor mired. Tractor *C* had a much larger supporting area, carrying less weight. The angle-lugs were spaced twice as far apart as on Tractor *B*. It tore out its footing and mired uniformly when exerting a pull of 2400 lb. This gave it 50 per cent more power in an emergency than Tractor *B*, though Tractor *C* was of lower rating. The spade teeth served a very useful purpose in catching and holding on submerged roots or timber in pulling out of a hole. The heavy skid-rings gave the driver control of his machine on soft ground.

This special equipment is not in any degree complicated or expensive. It easily can be made standard, to be attached to the tractor by the operator for soft-ground

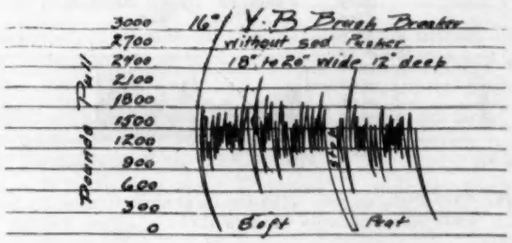

Fig. 16—Dynamometer Record of the Pull of the 16-In. *YB* Plow without a Sod Pusher

conditions. There are many tractors now owned throughout the country that cannot be operated profitably in the early spring or in wet weather. This is especially true in the working of peat land. Yet peat land can be cultivated, even at its wettest, without injury to the seed bed. It can be plowed in the spring and seeded, while the clay upland is still too soft. This is an extremely important factor in financing farming operations as well as in general farm efficiency. It seems logical that the wider wheel equipment be made standard for the wetter and softer districts and furnished upon the advice of the tractor dealer. It is also probable that, in many cases where tractors are now unprofitable on soft lands, an expenditure of $100 would change an unprofitable into a profitable investment.

Total Power Consumed

In calculating the power consumed per unit of cross-sectional area, the 16-in. plows give almost the same results as the 20-in. plows of the same make. This is not what was expected, as it was thought that the vertical cut would offer a higher proportional resistance than the bottom cut. Apparently, this was not the case, for

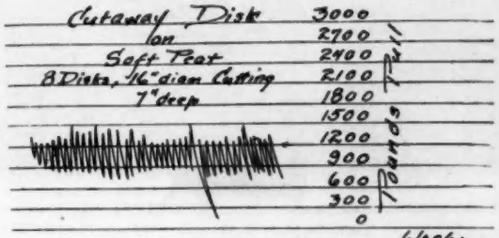

Fig. 17—The Record Made by the Dynamometer of the Pull of the Cutaway Disc Implement Equipped with Eight Discs When Cutting to a Depth of 7 In. in Soft Peat

the 20-in. plow gave no greater efficiency than the 16-in. plow of the same make.

There was an apparent advantage in efficiency in one make over the other. This is not thought to be due to any difference in the design of the plow bottoms themselves. In the case of the *XB* plow, however, the angle between the drawbar and the plow itself when

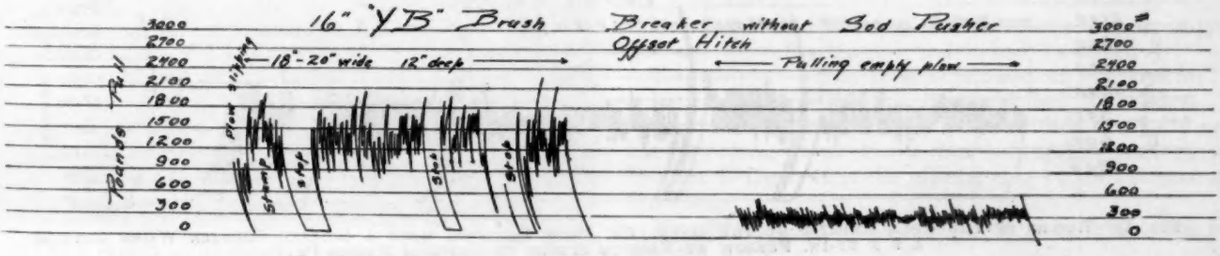

Fig. 14—The Pull of the 16-In. *YB* Plow Is Shown in This Reproduction of a Dynamometer Record

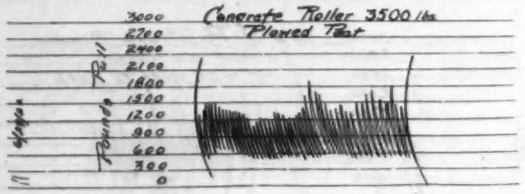

FIG. 18—THE PULL OF THE 3500-LB. CONCRETE ROLLER IS SHOWN IN THIS REPRODUCTION OF THE DYNAMOMETER RECORD

working was 18 deg. 20 min., while that of the *YM* and *YB* plows when working was 10 deg. 38 min. Of the dynamometer pulls on the two plows, the horizontal components parallel to the line of draft were, respectively, 5.33 and 3.95 lb. per. sq. in. of cross-section of the furrow slice. These figures compare favorably with the pull developed on the *YM* plow operating on this sodded peat with a direct pull, but bring out the great loss in power through friction and otherwise due to the angle of draft of the offset hitch.

The general proposition was established that for ordinary peat breaking, the horsepower required can be figured readily and with fair accuracy on the basis of a 5-lb. pull per sq. in. of the furrow slice. Multiply this by the speed it is proposed to travel and then calculate the horsepower in the ordinary way.

From the standpoint of mere efficiency in turning soil, the cutaway-disc implement loaded to cut 7 in. was far ahead of any of the plows. In fact, its power consumption per unit of cross-sectional area turned was less than one-half of that of the plows. Its use, therefore, is strongly to be recommended under conditions where its use is practicable. It cannot be recommended for first breaking instead of using plows, since it required about five passages of the disc-implement to give results

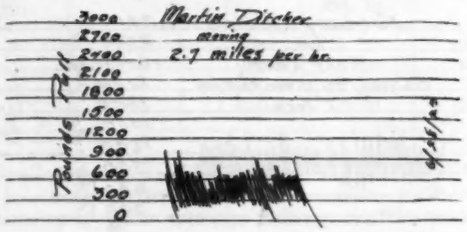

FIG. 19—THE DYNAMOMETER RECORD OF THE PULL OF THE MARTIN DITCH DIGGING MACHINE

equivalent to one plowing. The results of the work seem to indicate that the disc-implement may be extremely efficient for all later cultivation.

The roller gave some remarkable results. Not only did it pack the peat and so form a better root bed but, after the ground has been plowed, harrowed and rolled, even that part which was originally so soft and full of holes that it was difficult to get the tractors through became sufficiently firm to support horses for all later cultivation.

THE DISCUSSION

A. F. MOYER:—On what basis is the calculation made for power consumed per unit of cross-sectional area? Is it calculated on the width of the drive wheels and the amount of weight distributed on the drive wheels?

J. L. LARSON:—If we were to calculate the area and number of pounds per square inch on the drive wheels according to the depth they sank, the result would be about 2½ lb. per sq. in. for Tractor *C* in sinking 2 in. into the ground. A 1400-lb. horse will exert a ground

FIG. 21—A QUANTITY OF SOFT ROOTS THAT PASSED UNDER THE ROLLING COULTER AND PILED UP IN FRONT OF THE PLOW

pressure of about 15 lb. per sq. in., that of a man being about 6 lb. per sq. in. Tractor *C*, weighing 5000 lb. at a rough estimate, would exert a ground pressure of about 75 lb. per sq. in. The sinking into the ground would amount to 2½ lb. per sq. in.

G. R. B. ELLIOTT:—An interesting point brought out is in regard to the drawbar pull per square inch of cross-section of the furrow slice. When the furrow slice is too large or too small, the drawbar pull per square inch increases even though a decrease in the size of furrow slice may cause a decrease in the total drawbar horsepower.

GEORGE SYKES:—Would it not be better to increase the size of the roller?

MR. LARSON:—The concrete roller weighs about 3500 lb. and is very necessary on peat land. It is a question whether it would be practicable to increase that weight to any extent. We want to carry all the weight we can and yet have no slippage.

MR. MOYER:—Would that not depend on how the weight is distributed over the tractor?

MR. LARSON:—We found that to be a very important consideration. Many things must be taken into consideration when designing a tractor for soft land, and it is not as easy a proposition as it would seem to be.

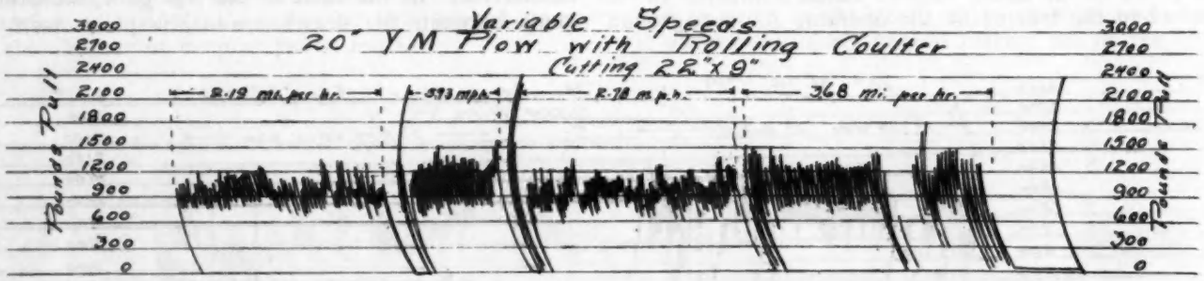

FIG. 20—RECORD OF THE PULL EXERTED BY THE 20-IN. *YM* PLOW EQUIPPED WITH A ROLLING COULTER WHEN CUTTING A 9 X 22-IN. FURROW AT VARIOUS SPEEDS IN UNIFORM SODDED PEAT

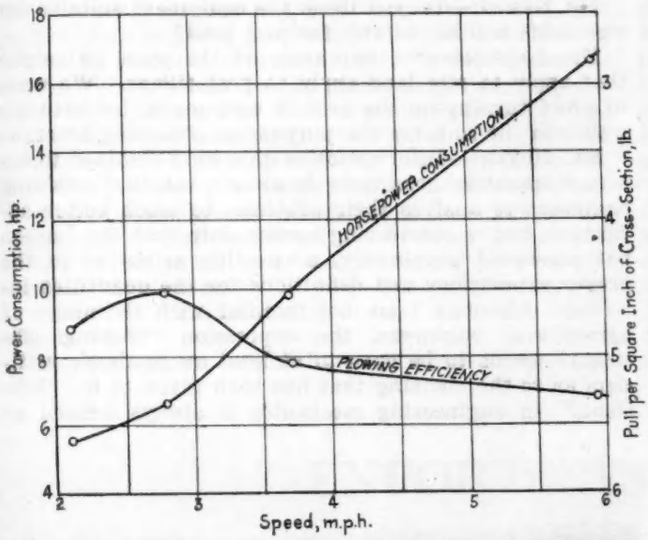

FIG. 22—CHART OF THE PLOWING EFFICIENCY AND THE POWER CONSUMED BY THE 20-IN. MARSH PLOW WHEN WORKING IN PEAT AT VARIOUS SPEEDS

MR. MOYER:—Have you in your observations, not necessarily in your measurements, observed the effect of the wheel diameter?

MR. LARSON:—We found the larger wheel-diameter much better than the narrow or small wheel.

MR. SYKES:—Why is the roller made so small in diameter?

MR. LARSON:—The main reason is that we used the only available steel casing we had.

MR. ELLIOTT:—We wanted to get as near as possible to a weight of 1000 lb. per linear ft. of width of the roller.

MR. MOYER:—Is that at only one speed?

MR. LARSON:—That was taken at 2¾ m.p.h.

MR. SYKES:—Would it not be of advantage to have a larger diameter for the roller?

MR. LARSON:—Not necessarily. We want to keep the weight down and still pack the ground sufficiently hard. We think the roller we used does this.

MR. MOYER:—How do you express plowing efficiency?

MR. LARSON:—By the number of pounds required per square inch of furrow slice.

W. G. CLARK:—Was this work purely experimental?

MR. LARSON:—Yes.

MR. CLARK:—Have you any basis for stating the cost of doing this work?

MR. LARSON:—Not yet, but we did much more work on this plot of ground than we think would be necessary for the farmer. I doubt if the farmer can afford the expense. Most of it would be done by the farmer at the time of the first breaking. However, it is absolutely necessary that he does some of the things we did.

MR. CLARK:—It would need to be cheap land.

MR. LARSON:—The land needs to be cheap and it is cheap. Within the next 5 years it will be surprising how much of this land is plowed up. It raises a very good crop; in fact, our best seed potatoes come from peat land.

MR. CLARK:—What other crops are particularly adapted to peat land? Will the crop pay the expense?

MR. ELLIOTT:—About two-thirds to three-quarters of the ordinary farm produce will do better on peat land properly cultivated than on up-land; the remainder will not do so well. Corn, potatoes and rye do especially well, but wheat will not do anything. Oats in this part of the Country will produce nearly twice as much. Nearly all grasses and clover for hay do excellently well. Most truck and garden crops do especially well. Onions should range from 500 to 600 bu. per acre and I know of one crop of more than 1100 bu. to the acre. Carrots returned us 600 bu. to the acre. Both of these were grown where the water was held 6 ft. below the surface. Sugar beets gave 20 tons to the acre, yielding 17.8 per cent of sugar. Where the water is held near the surface, root vegetables are apt to "finger" badly.

FIG. 23—THE MARSH OR *M* TYPE OF PLOWSHARE EQUIPPED WITH A FURROW PUSHER

FIG. 24—HOW THE FURROW PUSHER OPERATES TO PUSH THE FRESHLY TURNED FURROW TO THE RIGHT SO THAT THE SUCCEEDING FURROW WILL FALL FLAT AND BE PUSHED TIGHTLY AGAINST IT, THUS COVERING THE GRASS AND OTHER LITTER COMPLETELY

A. H. BATES:—You mentioned an 18-deg. angle of hitch.

MR. LARSON:—One is 18 deg. and the other is 10 deg. The 18-deg.-angle hitch is on the *XB* plow and the 10-deg.-angle hitch is on the *YM* plow. With this angle of side-draft, it gave approximately 150 lb. more pull than on a straight draft.

MR. MOYER:—What effect is caused by the plow having side-draft?

MR. LARSON:—We did not have the opportunity to measure it on the tractor, but it had its effect.

MR. MOYER:—Was it less marked on the tractor?

MR. LARSON:—I think it would be. Of course, the tractors had sufficient excess power. Side-draft did cause hard steering at all times, but it was less serious with Tractor *C* than with the others, due to its extreme narrowness.

MR. SYKES:—At what speed do you obtain the most efficient work?

MR. LARSON:—At a plowing speed of about 2¾ m.p.h.

FIG. 25—THE 3500-LB. CONCRETE ROLLER EMPLOYED TO PACK THE SEED BED

MR. MOYER:—Do you disregard the number of acres covered per day?

MR. LARSON:—Yes, so far as present calculations have gone.

MR. SYKES:—At a plowing speed of 1¾ m.p.h. would the equipment sink in the bog?

MR. LARSON:—Yes, the possibility of miring is doubled by running at slow speed; we can plow better and plow more ground at the 2¾-m.p.h. speed.

MR. MOYER:—Have others had a similar experience on other types of land? In field experiments, I observed that the resistance will usually increase as the speed increases.

MR. LARSON:—The surface of a bog is like rubber or tar, the faster the equipment moves over it, the less impression it makes.

MR. MOYER:—Time is an element in depression and I

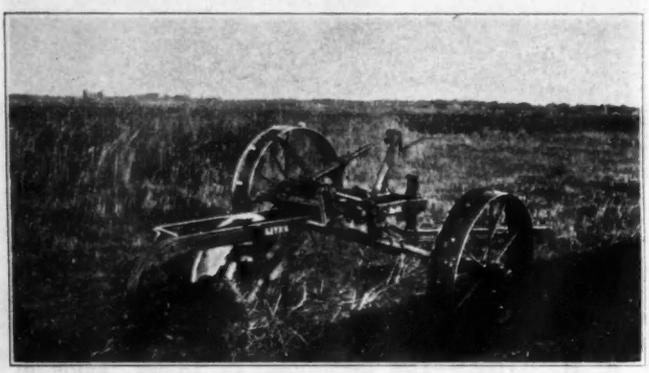

FIG. 26—THE TYPE *M* PLOW EQUIPPED WITH A ROLLING COULTER AND WORKING IN THIN SODDED PEAT

The Clean Cutting without Clogging by the Roots Should Be Noticed

presume that, with a higher speed for a short length of time, the equipment would not sink so deep.

MR. LARSON:—Our experiments seem to indicate that statement to be true, so far as miring goes.

MR. MOYER:—A bog has a certain resilience.

MR. SYKES:—Are the lugs on the drive wheels used as a result of special experiment, or are they standard?

MR. LARSON:—They were put on specially but are standard. Up to about 3 years ago some of the same equipment was not standard, but this equipment had been made standard by at least one company when we carried on our experiments. Tractor builders who put their tractors in the rice fields gave it considerable thought, and I think it will be the same with regard to peat land.

MR. SYKES:—Do you think the equipment suitable for rice fields will be suitable for peat land?

MR. LARSON:—No, but some of the same principles that apply to rice land apply to peat tillage. We want to avoid turning-up the ground very much, but still get well into the soil, for the purpose of obtaining traction.

MR. MOYER:—Information of this kind obtained under actual operating conditions is always valuable, enabling engineers to analyze their problems to much better advantage, but in considering certain data that Mr. Larson has presented graphically, a question arises as to the proper expressions and definitions for the quantities involved. Although I am not familiar with the usage of agricultural engineers, the expression "plowing efficiency" seems to be open to at least an academic criticism as to the meaning that has been given to it. "Efficiency" in engineering mechanics is always defined as

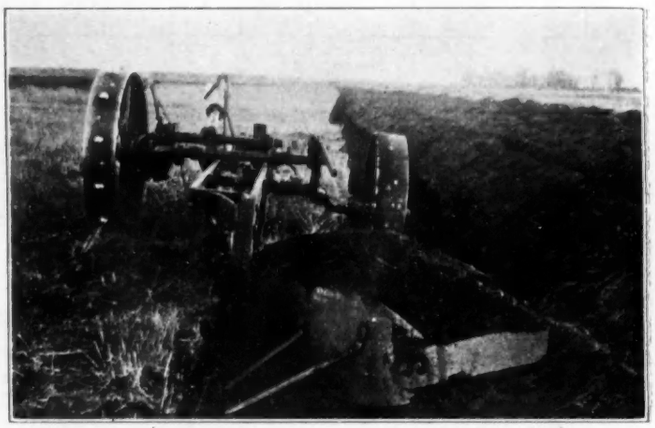

FIG. 27—REAR VIEW OF THE *YM* PLOW EQUIPPED WITH SOD PUSHER THAT WAS ILLUSTRATED IN FIG. 9 TURNING A 9 X 22-IN. FURROW

the output divided by the input and usually involves quantities of energy expended and work accomplished. Its denomination is usually in percentage, but it is expressed in square inches per pound in the relation between the furrow section and the drawbar pull that Mr. Larson has plotted as "plowing efficiency." Would it not be better to invert this quantity and call it "specific draft," "unit draft" or some such more accurate name expressed in pounds of draft per square inch of furrow section? It is certainly an important quantity to consider in studying the draft of plows and should be given a name that is accurately descriptive.

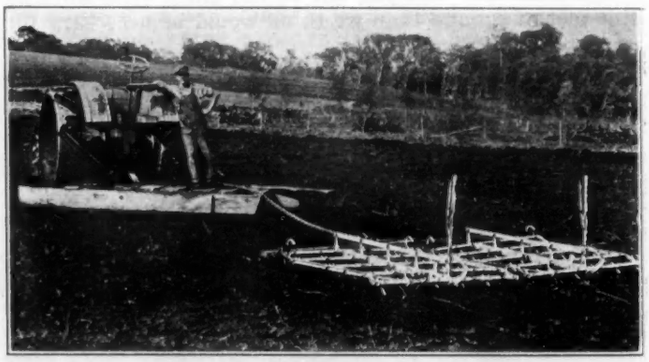

FIG. 28—HOW THE LAND LOOKED AFTER THE PLOWING WAS COMPLETED AND HAD BEEN GONE OVER WITH A GROUND SMOOTHING ATTACHMENT FOLLOWED BY A HARROW

Mr. Larson has used the qualifying words that this quantity denotes plowing efficiency "without considering the number of acres plowed per day," which raises the question: What would be the plowing efficiency if one does consider the number of acres plowed per day? And not only should we consider the number of acres per day, but the depth of plowing as well. Returning to the definition of efficiency, it seems that we might include all of these considerations by calculating some such quantity as cubic feet of furrow turned per drawbar horsepower per hour. This would be a true measure of plowing efficiency and is very nearly in proportion to plowing cost when depth is considered and human labor per horsepower is fixed.

If the data shown in Mr. Larson's curve of "plowing efficiency" or, preferably, "specific draft" were recalculated according to the above proposed definition of plowing efficiency, there would no longer be a "peak" in the revised graph at 2¾ m.p.h., but only a slight flattening of the curve as the speed is increased, showing that the gain in efficiency is less rapid beyond that speed; but a gain is nevertheless available up to speeds much higher than those commonly used. Since it is evident that this difference in viewpoint will show a rising efficiency with an increased speed in certain ranges where a falling efficiency has been shown, it is important to be specific in our definitions. It would be extremely interesting if Mr. Larson would reply to this discussion and, if he sees fit, prepare a curve of plowing efficiency as above defined, referred to speed.

EXPORTS AND IMPORTS

IT has been known to students of the subjects for a good many months past that far-reaching changes of proportions and relations were in progress in the commercial (and financial) interchanges of the United States with the rest of the world. It had become clear before the fiscal year 1922-1923 had far advanced that important shifts of emphasis were occurring on both the import and the export side of the account. The most impressive of these, of course, was the remarkable growth of the value of the imports in proportion to the value of the exports, particularly when the large net balance of imports over exports of gold and silver was taken into the reckoning. Indeed, the fiscal year had run scarcely half its course before it began to appear possible that for the first time in nearly three-quarters of a century we should in 1922-1923 as a whole have an excess of imports rather than exports in our total visible interchanges with other nations. Particularly striking though this great change was, however, it was by no means the only shift of economic significance that began to reveal itself in our import and export figures soon after the last fiscal year began. It was early observable that the values of our imports of crude materials for use in manufacturing and of manufactures for further use in manufacturing were swelling very rapidly, while the value of our imports of manufactures ready for consumption was not increasing at anything like the rate that had been predicted by many protectionists or that might naturally have been expected in view of the extensive rise of the price level in the United States. On the other hand, the export figures, when analyzed showed a progressive increase in the proportion borne by the value of the exports of products of manufacture to the total value of the exports, accompanied by a progressive proportional decrease of the value of the exports of foodstuffs and raw materials. In other words, as the past fiscal year progressed the foreign trade of the United States came more and more to present a general appearance similar to that normally presented before the war by the trade of the leading industrial nations of Europe—above all Great Britain. Like those nations in the pre-war years we had what is commonly, though unscientifically, described as an "unfavorable" or "adverse" balance of trade in our total visible interchanges with the rest of the world, and at the same time our manufacturing industries were playing an ever larger part in determining the inflow and outflow of commodities and goods.

On our total exchanges of merchandise with other countries during the first 11 months of 1922-1923 there was a "favorable" balance; that is, an excess of the value of the exports over that of the imports, in the amount of $175,315,362, whereas in the first 11 months of 1921-1922 the "favorable" balance on merchandise account was the very much larger amount of $1,088,421,629. The latter of these figures fairly corresponds, if due allowance is made for changes in the level of prices, with the normal "favorable" balance shown by the trade of the United States in the pre-war years; but evidently the figure for 1922-1923 represents but a mere fraction of the "favorable" balance to which we were formerly accustomed. Moreover, we by no means get the whole story of the change that has occurred from the figures of our imports and exports of merchandise. Besides these the values of our imports and exports of gold and silver must be reckoned with. The total inward and outward movement of the precious metals during the period of this year yielded an excess of imports in the amount of $221,936,550, against an excess of imports of the precious metals totaling $437,245,407 for the period a year ago. Here the important thing to note is that when we set the 1922-1923 net import balance on account of gold and silver over against the year's net export balance on merchandise account we are left with a final net import balance in the amount of $46,621,188 on the Country's total "visible" interchanges with the rest of the world during the 11 months' period, whereas a year ago, notwithstanding the considerably larger net importation of the precious metals, our total "visible" interchanges with other countries still showed an excess of exports over imports in the amount of $651,176,222, a "favorable" balance far smaller, of course, than those which were usual from year to year during and immediately after the war, but nevertheless on its face normal enough by the pre-war standard.

The two groups or classes of imports which made really large gains in value in 1922-1923, as compared with 1921-1922, were crude materials for use in manufacturing and manufactures for further use in manufacturing, while the group designated as manufactures ready for consumption increased only insignificantly. The figures undeniably indicate a remarkable state of activity and prosperity among our industries, necessitating the importation of raw material and semi-finished products upon a quite unprecedented scale. The change in the conditions of the Country's foreign trade during the past fiscal year was not attended by a diminution of the ability of our industries to compete with those of other countries.—A. R. Marsh in *Economic World.*

The Testing of Materials and Its Effect on Engineering

By G. B. Upton[1]

Buffalo Section Paper

THE development of the automobile has brought with it new materials and new methods of control of material and has altered machine design by emphasizing the importance of allowing for fatigue, stiffness and shock-resistance as coordinate in importance with strength. Progress has been concerned chiefly with the construction and operation of the engine, with the methods of manufacturing the parts and with the properties of the materials of which the parts are composed. Inseparable from quantity production is the system of gaging and inspecting the size and shape of every part manufactured. Another kind of inspection is that of checking the properties of the materials.

The former method of testing a test-piece to destruction has been superseded by that of testing every finished part without injury being done to the part. Strengths and working stresses now are commonly as high as five times those of a generation ago. Machinery is correspondingly lighter, the range of operation wider and the cost of operation lower. Light alloys of aluminum or magnesium base, unknown a generation ago, have come into use, and a new cast-iron with three times the strength of ordinary gray cast-iron and having some ductility and very appreciable shock resistance has been produced. Elastic-limit, in its former sense, soon will be relegated to an unimportant place and, for it, will be substituted the broader term "fatigue failure"; an "endurance curve," plotted as the logarithm of stress against the logarithm of the number of cycles needed to break, will show the relations in a perfect piece by a nearly straight line. Internal uncleanliness and external roughness will increase the downward slope of the curve. No definite correlation seems to exist between the fatigue elastic-limit and the proportional elastic-limit or the yield-point.

In some of the copper-rich alloys, the fatigue elastic-limit probably will be found to be higher than either the proportional elastic-limit or the yield-point. Among non-ferrous metals the fatigue limits are low. Tests of duralumin should be made. Through such researches as those of Professor Coker in England, who makes stresses measure themselves by polarized light passing through a loaded body of transparent material, the stresses at fillets, changes of section, keyways, scratches and the like are coming to be appreciated better. The weakening of a piece by a scratch or a groove has been found to be a function not only of the ratio of the depth of the scratch to the radius at the bottom of it, but also of the ratio of the depth of the scratch to the thickness of the piece. A scratch or a groove becomes relatively more serious, the smaller the piece is. Endurance curves of sheet metal are affected greatly by surface imperfections that are relatively large compared with the thickness of the piece.

Alloy-steels can match the light metals in the ratio of strength to weight for tension loading only. A generation ago, strength was the controlling factor in design; whereas stiffness should have been given first place. At present, the four coordinate bases of passenger-car design and of choice of material are, in their order of importance, stiffness, fatigue-resistance, strength and shock-resistance. In truck work, strength comes higher. The particular requirements of each part should be investigated. In many parts in which stiffness is essential, such as the crankshaft, case-hardened low-carbon steel would give adequate strength and hardness and would be cheaper, easier to forge, machine and heat-treat than would alloy-steel. Strength should not be the only factor of safety considered in design; there should be as many factors of safety as there are bases of design; strength, fatigue, stiffness, shock.

ENGINEERS who are interested in the progress of their profession will find the story of the automobile during the last 20 years fascinating. The automobile has brought with it and has put into commercial practice new materials, new methods of control of material, especially in alloy-steel and the light metals, and the heat-treatment of metals for high strength; has altered machine design by emphasizing the importance of allowing for fatigue and for stiffness or shock-resistance, as being coordinate with the design for strength; and has made obvious the necessity for considering inertia-stresses in the design of moving machinery. The scientific design of bearings and lubrication will perhaps be the next boon to general engineering knowledge to come through the automobile; it has not yet arrived, but is pending.

There have been three parallel lines of development in the progress of automotive engineering; (*a*) the theory and practice of the construction and operation of the prime-mover, or engine, used; (*b*) the improvement of the manufacturing methods for producing and shaping the materials of construction, and for assuring the desired shape, size and proportion of different parts; and (*c*) the radical and important improvement of the properties of the materials of construction that makes possible the most astounding reductions in the sizes and the weights of parts.

The improvement of production machinery, such as forges, presses, lathes and milling, drilling and grinding machines, that makes possible the modern system of "interchangeable" quantity-production, is in itself a wonderful piece of engineering work. Inseparable from it is the system of gaging and inspecting the size and shape of every piece manufactured. Another kind of inspection, upon which we have made probably only a beginning, is the check on the properties of the materials, a discussion of which comes properly under the title of this paper.

Not so very long ago, material inspection was done mainly by tests to destruction of an occasional piece that was shunted out of production. A certain fraction of the current product was set aside for such tests, usually 1 piece in 100 or 1000, but a sufficient number to be considered typical of a run, a "heat" from a steel-furnace or a similar quantity. If the test-specimens that were selected passed the destruction test, all their companion pieces were accepted; if the test-specimens failed, further

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tests were made to check the first results and to decide upon the acceptance or the rejection of the lot. At present, though such testing to destruction of selected pieces has not been abandoned, this scheme belongs rather to the preliminary research stage; and, in the course of production, the checking of material is done best by tests that can be applied to every finished part without injury being done to the part. Future progress in the control of material will lie largely in this inspection; for example, the present Brinell and scleroscopic tests of hardness are unlikely to survive the next 10 years as, in practice, both have too many obvious defects both of theory and of performance.

For a broad and detailed discussion of the relation of the engineer to materials and material-testing, I recommend a recent book on Engineering Steels, by the English engineer, Leslie Aitchison. He points out that the demands of the automotive industry for steels capable of withstanding the higher stresses have made it imperative for engineers to ascertain, before using them, that the steels really possess the expected and desired properties; hence, there is a critical re-examination of what are, fundamentally, the properties the engineer needs for specified purposes, and a development of better means for measuring those properties. Specifications are often broadened to cover the chemical composition, heat-treatment, macrostructure and microstructure, as well as the physical properties, strength, ductility, hardness and the like. When we begin to broaden specifications in this way in attempting to tie the case up tight, to obtain absolute assurance that we shall get what we want, we admit that we do not know exactly what we do want, or how to state that want simply and in measurable terms. We should be able to make a specification in terms of three measurable properties, certainly not over four, and to state the tolerance limits for each so that the finished material shall be exactly what we want without restriction as to how it is made or what it is made from. We do not now have such a knowledge of materials. At present we need improvement in our knowledge of the properties of materials rather than improvement in the properties themselves.

Comment on the actual improvements in the properties during what may be called recent times is hardly in point here, as the past and current values are so well known. Our fathers worked with structural and machine steels of carbon-steel composition, and used them, in general, without heat-treatment. We use heat-treatment and alloy-steels very commonly and get strengths and working stresses of an order as high as five times those of a generation ago. Correspondingly, our automotive machinery is lighter where weight was formerly required; and, in consequence of lower weight, the range of operation has been widened and the cost of operation lowered. As L. H. Pomeroy recently has pointed out, a saving of weight in one place has cumulative effects throughout the car. We also have a new set of materials with which to deal in the light alloys of aluminum or magnesium base, which were unknown a generation ago, and the heat-treatment of them promises to be as important as the heat-treatment of steel.

In the field of cast iron, new material should be evolved in the near future. In the course of experiments on the quick conversion of white cast-iron into malleable cast-iron, it has been found possible, by a process taking but a few hours of heat-treatment, to make a material having a gray fracture due to "temper graphite," with a tension yield-point of about 45,000 lb. per sq. in., a maximum strength of about 75,000 lb. per sq. in. and some ductility. This new material is practically a gray cast-iron of about three times the strength of ordinary gray cast-iron, with some ductility, and with a very appreciable shock-resistance. It seems well worth investigating as a material for parts such as timing gears, inserted cylinder-sleeves and the like and, in general, for parts of not too complicated section, for which a cast iron of higher strength and stiffness than those of gray iron might be usable. This new material should be cheaper than malleable cast-iron.

Properties of Materials

In studying the properties of materials and learning what we should measure and specify as significant properties, it seems that the elastic-limit, as "proportional limit," or "Johnson limit," or the like, is soon to be set back to a very unimportant place. This property is not strictly a fundamental one; it is an accident of the manufacturing process, or largely of the particular history of the piece. We are interested in elastic-limit in two ways. If a part changes shape under service stresses, normal or abnormal, with a permanent set that is greater than the limits of the accuracy of machining and assembling, the part ceases to function properly with the adjacent parts. This furnishes a rational basis for a specification of one practical kind of elastic-limit, the stress that first causes a permanent set of an arbitrary value, such as 1 part in 10,000. Beyond this we may well recognize a yield-point, or yield-stress, at which the permanent set rapidly increases at an intolerable rate. An arbitrary definition might give this as the stress that causes the permanent deformation to pass the relative deformation by 0.005, or ½ of 1 per cent. Definitions such as these would have direct practical engineering application, would be fairly easy to use in testing and could be applied alike to all materials, ferrous or non-ferrous, whether heat-treated or not.

Our second interest in elastic-limits is in a field of the highest importance, which now for the first time is receiving a study that is detailed, accurate and broad; this is, the field of "fatigue failure." For ferrous metals, a large quantity of data on the investigation of fatigue has been obtained at the University of Illinois Engineering Experiment Station, through the cooperation of this station with the National Research Council, the Engineering Foundation and the General Electric Co. Many other sources have contributed information, and a grouping and a correlating of all the data and test methods are being carried out at the Bureau of Mines, by H. W. Gillette.

It is clear that for each steel, when it is clean and sound, of perfect surface smoothness and shaped carefully to avoid local overstresses, there is a limiting elastic-stress range that can be repeated a more or less indefinite number of times without the failure of the piece. If the repeated stress is the least bit too high the endurance of the steel, the number of stress repetitions for fracture becomes definitely finite; the repetitions become fewer as the stress becomes higher. The endurance curve, plotted as the logarithm of the stress against the logarithm of the number of cycles needed to break, consists of a straight or nearly straight line that comes down somewhat rapidly from the high stresses and small number of cycles to the elastic-stress and a cycle number somewhere in the region between 500,000 and 5,000,000; thence, the curve seems to be flat and of constant stress. But this is for the perfect piece. When the steel is not clean and sound, the whole curve is lowered and may become very erratic. When the surface is not

smooth, but contains even tool or grinder marks crossing at angles the lines of a repeating tension-stress, the stress and endurance are lowered. Even the shape of the endurance-curve changes; its second branch no longer goes to infinite durability at constant stress, but slopes downward, and it slopes the more sharply the greater the internal uncleanliness or the external roughness is. Fillets or curved surfaces evidently cause much higher real stresses to occur than are commonly imputed to them.

Endurance Curves

The second kind of elastic-limit is that of the endurance curves. Means of finding this limiting-stress quickly and reliably are now being evolved, and also means of inferring its probable value from other more common testing values, such as the Brinell hardness-number, the tension at maximum strength and the like. No definite correlation seems to exist between the fatigue elastic-limit and the proportional elastic-limits or the yield-points. In some cases, the fatigue limit seems to be higher even than the yield-point. We have substantially no information yet on the fatigue properties of non-ferrous metals. It seems probable that, for some of the copper-rich alloys, the fatigue elastic-limit will be found to be very much higher than their proportional elastic-limits, and perhaps higher than their yield-points. This needs to be proved; for some of the non-ferrous metals we may expect to find the fatigue-limits very low indeed. There is a chance that an investigation of the fatigue properties of copper alloys may be financed soon by the Engineering Council and an association of the manufacturers of copper and brass. No one seems to volunteer to finance the fatigue research of the light metals. The use of duralumin connecting-rods suggests that we should find out the fatigue properties soon either by test or by experience. The stressing of the connecting-rod, by reversing the tension and the compression, is harder on the material than reversing bending or torsion, because it stresses the central parts of the section where inclusions, segregations, pipes, flaws and weaknesses generally are most likely to occur.

It might be mentioned that the fatigue-limit in torsion has, as it should have, a shear-stress of a numerical value one-half the tension-compression value that is found in reversing bending. It seems probable that all cast metals, even steel, will have no true fatigue elastic-limits, but will have endurance curves continually dropping to lower stresses as the "life" is increased. The fatigue-failure differences between the cast and the worked metals have not been investigated as yet.

Partly from fatigue researches, but more from such researches as those of Prof. E. G. Coker in England, who makes stresses measure themselves by polarized light passing through a loaded body of transparent material, we are coming to a better appreciation and nearly to a measurement of the stresses at fillets, at changes of section, at roots of keyways, at scratches, and the like. And, as we apply this knowledge to practice, many of the details of design will change. For example, a groove like a keyway in a shaft, of a depth x and a bottom radius r, causes a local enhancement of the shear-stress dependent on the ratio x/r. If this grove lay squarely across the direction of a tension-stress, the enhancement would be more serious. For a splined-axle shaft, a radius at the bottom angle of the splines of 0.1 of the depth will bring the enhanced stress to 2.3 times the normal stress; a radius of 0.2 of the depth will make the stress only 1.5 times the normal stress. We all are familiar with splined-shaft breaks that start from the angles at the bottom of the splines. Of course, truly sharp corners and theoretically infinite stress do not occur; but the sharp cracks from excessive cooling-stresses in heat-treatment operations are very serious. Such cracks will cause an early fatigue failure if the part goes into service. On a single overload, a sharp corner will run the local stresses up beyond the yield-point; the yielding then rounds the corner and stops the further rise of the stress.

In practice, the weakening of a piece by a scratch or groove, by stressing in bending or in torsion, is a function not only of the ratio of the depth of the scratch to the radius at the bottom of it, but also of the ratio of the depth of the scratch to the thickness of the piece. A scratch or groove of given dimensions is relatively more serious the smaller the piece is. Endurance curves made on sheet metal, such as those of the recent McCook Field tests on sheet duralumin, must be affected greatly by the consideration of surface imperfections that are relatively large compared with the thickness of the piece. McCook Field has found that for duralumin sheet there is no fatigue elastic-limit stress, up to at least 500,000,000 cycles. It is possible that this finding is not inconsistent with the presence of a definite and reasonably high fatigue elastic-limit in forged duralumin connecting-rods. I write this paragraph largely to show our real lack of knowledge of the fatigue action of the light metals, knowledge that we need before we can go extensively into any complete substitution of light metals for steel in parts carrying transverse or torsional or long-column compression loading, and where the moment of inertia of the section is a large factor in the strength of the part. Extensive use of light metals is undoubtedly in prospect if the fatigue properties under the conditions both of casting and of forging are found to be satisfactory. Alloy-steels can match the light metals in the ratio of strength to weight for tension loading only.

What Engineers Design For

This brings us back to the question of what determines the choice of materials, and what, in general, are the conditions of design; that is, what we design for. I am not discussing costs. We inherited from our predecessors and our textbooks are filled with a scheme of design having strength as the controlling factor. The design for strength grew up in an age when the speed of moving parts was low, so that inertia forces were only of secondary, not primary, importance; and fatigue-failure was an improbable rather than an ever-present possibility. Even in that day the design of machine-tools, such as lathes and shaping, planing and milling machines, should have been based first on stiffness, and only secondarily on strength, for the first duty of a machine-tool is to hold the cutters rigidly in place with regard to the work.

In the automobile business we know, though we do not always practice what we know, that there are four coordinate bases of design, and from design, of choice of construction material. These four are strength, fatigue-resistance, stiffness and shock-resistance. In importance they may rank, for passenger-car design, as stiffness, fatigue-resistance, strength and shock-resistance. In truck work strength comes higher in the list. Design for shock-resistance in some respects is hardly within our province; it is not entirely reasonable that an automobile should be able to survive a collision with a stone wall. But it should be able to survive a clutch dropped into engagement on a "racing" engine, a case that conditions the strength and stiffness of every gear and shaft from the clutch to the rear wheels, for then the loadings, on

account of the acceleration forces of the rear wheels, are much higher than are needed to slip those wheels in more nearly normal operation.

An engine crankshaft furnishes a good example of the shift of the basis of design. At first it was strength, then fatigue; now, it is stiffness, to resist the torsional vibration. Each change of the basis of design brought an increase of the dimensions of the shaft compared with those of the cylinder-bore. And in this we do not practise as we preach, or as we know. For, in a general way, strength, against either single overloads or fatigue-failures, goes up with the tensile-strength of the material and the ratio of the moment of inertia to the diameter of the shaft, while the stiffness depends on the product of the modulus of elasticity of the material and the moment of inertia of the shaft. When we designed for strength alone we kept shaft diameters small and used nickel-chrome heat-treated steel for high strength and good hardness. When we design for stiffness, such a material is too good. A low-carbon steel would be just as stiff, for it has substantially the same modulus of elasticity; and, when the diameter is large enough for satisfactory stiffness, a case-hardened low-carbon steel would give adequate strength and hardness to the crankshaft and be cheaper and easier to forge, machine and heat-treat. But, lacking the courage of our convictions, we keep on using alloy-steel. So we could go through the parts of a car, examining everywhere the basis of design and the choice of materials. Steering-knuckles are controlled by fatigue-resistance, front axles by strength and shock-resistance, propeller and axle-shafting by strength and shock-resistance, gears by strength and wear-hardness, crankcases by stiffness, and so on.

The frame is very interesting. For a passenger car, the dominant property needed is stiffness, so that the frame and the body may agree with each other as the car runs. A partial realization of this is shown in the great deepening of the frames in recent years. For the most part, this deepening misses the point. It increases the bending stiffness that was never seriously bad anyway. The frame is nearly a two-dimensional rectangular part, loaded at the four corners by the springs and along the long edges by the body and the engine. Its defect of stiffness is against weaving or torsion, a twist of one end with regard to the other. Deepening of the side-rails does not make the frame torsionally stiff. We may find a way out by removing the cause of the weaving, by changing the spring loading to three points instead of four, by using transverse springing at one end of the car. If we stick to four-point springing we must meet the torsional forces with torsional rigidity of structure, which requires a structure in three dimensions instead of two. We might take a hint from the fuselage of an airplane, and make a permanent-top body a unit with the frame, making this unit simultaneously much stiffer and much lighter than the present structure of two parts quarreling with each other. The new "frameless" Lancia car shows a start in this direction. A mild low-carbon steel with be just as stiff as an alloy-steel; so that, as long as stiffness controls frame design, there will be nothing to gain from alloy-steel.

Factor of Safety

So long as design was based solely on strength, the old definition, "factor of safety equals the maximum strength divided by the working load," was all right. Then the argument was made that the yield-point, rather than the tension maximum load, should be used in calculation. This argument originated in the fact that the heat-treated carbon or alloy-steels have yield-points relatively much nearer to the maximum than the familiar 50 per cent relation of the normal carbon-steels. In fatigue research we find the fatigue elastic-limit to be substantially 50 per cent of the maximum strength in all worked steels, regardless of their analysis or heat-treatment. This seems to throw us back to the old scheme of using the maximum strength, rather than the yield-point, in computing the strength factor of safety. We should not forget that for long columns in compression, for I-beams or channels in bending or for tubing under torsion, the yield load is substantially the maximum load.

There are other factors of safety than that for strength to be considered, factors for deflection or rigidity, shock resistance, elastic-limit, break and the like. The numbers designating these factors of safety may be, and usually are, very different from the strength factors. Take resilience or shock-resistance as an example, in a steel with a yield-point of 50,000 lb. per sq. in.; maximum strength, 100,000 lb. per sq. in.; ductility, 30 per cent. Let the working-stress be 15,000 lb. per sq. in. Then the ratio of the strength factor of safety to the yield point is 3, to break is 6; the resilience factor to yield-point is $(50{,}000/15{,}000)^2 = 10$, and to break is about $20{,}000/4 = 5000$. The elastic work per cubic inch at working stress is $50{,}000^2/(2 \times 30{,}000{,}000) = 4$ in-lb.

Summarizing, the study of factor of safety brings up the same argument we have had before: The basis of design cannot be merely for strength. Hence, we cannot state a single factor of safety, but must have as many as there are bases of design; strength, fatigue, stiffness, shock.

THE DISCUSSION

Otto M. Burkhardt:—It is a problem to take a successful car of today and, without changing the design completely, to eliminate, let us say, 100, 200 or even 500 lb. of weight without imperiling the product in one way or another. The high-speed prime-mover is looked on as offering some solution. A precedent is furnished by the steam engine, with which we have noted that every time a new type of engine has been offered, the speed has been increased. The advantages of the high-speed engine are well known, and so are the disadvantages. There is much room for the exact analysis of stresses and deflections and, when the opportunity arrives, there will be modifications in favor of lighter weight.

John Miller:—What does Professor Upton think of the use of case-hardened steel for resisting fatigue? I have observed that case-hardened steel seldom breaks on account of fatigue-failure, and I have found that case-hardened alloy-steel will outlast plain alloy-steel in the Stanton repeated-impact test. In a passenger car or a motor truck, we sometimes get stresses as high as those in the Stanton test. In case-hardened steel we get a very high elastic-limit on the surface, where it is most needed, and there appears to be an absence of internal stresses; whereas, in high-carbon steel of the same hardness, the useful strength of the steel is reduced by the internal stresses.

Prof. G. B. Upton:—If not hardened above 400 Brinell, the fatigue elastic-limit should go up with the carbon content of a steel; and Mr. Miller's experience is in line with what we should expect for case-hardened stock under bending or twisting loads. I have had no experience in this case.

Mr. Miller:—I agree with Professor Upton that the light car is the coming car, and I believe that within the

next 5 years we shall be building cars of about one-half the present weight.

PROFESSOR UPTON:—The trouble is that the light car is a harder car to design.

MR. BURKHARDT:—Professor Upton spoke of following airplane design in making the top permanent with the body, so that the weight of the car could be cut down and the stiffness increased at the same time. Still, some persons like to ride with the top down. This improves the appearance of the vehicle and of the occupants as well. The trouble is that engineers are dictated to by the public. The permanent top would be a fine thing, but the public must be served.

DONALD COX:—What is the relation between the fatigue elastic-limit and the tensile-strength?

PROFESSOR UPTON:—Roughly speaking, the fatigue elastic-limit is about one-half the tensile-strength. This information can be obtained from the special bulletins issued by the University of Illinois.

MR. COX:—Ordinarily, then, you can say that the tensile-strength is approximately 500 times the Brinell hardness number?

PROFESSOR UPTON:—Yes. You probably will find that it has been worked out more perfectly for the carbon steels. It depends on whether the fatigue stressing is symmetric about zero stress. If it is unsymmetric, the permissible range is less. Because the high-speed engine has a smaller combustion-chamber, the compression pressure can be higher. Racing engines with a 2¾-in. bore have to run at 4000 r.p.m., and they do not "pink" with a compression-ratio of 7 to 1.

A. J. FITZGIBBONS:—Why is there such a wide difference in methods of design between the Society and the American Society of Civil Engineers? One is decreasing the weight and the other increasing it.

PROFESSOR UPTON:—The civil engineers have reached the limit of weight and are reducing it. Rails are being made lighter, as are also the wheel loads.

MR. BURKHARDT:—Locomotives require adhesion and therefore need weight. The railroads are not so successful on the short haul. As automobiles have to accelerate and retard between city blocks and a locomotive does not, the difference becomes apparent.

DAVID FERGUSSON:—The question of designing for stiffness is of much moment today and we are using, in many places, expensive alloy-steels that there is now no reason for adopting. In the past, to overcome the breaking of some parts such as engine crankshafts, that were made of carbon-steel, we changed to chrome-nickel steel. This was logical at the time. On the other hand, we have been required to make engines run so much more quietly since then, that we have had to use crankshafts of much larger diameter to eliminate torsional vibration; and plain carbon-steel should give satisfaction with these increased dimensions. The only advantage in these larger crankshafts of chrome-nickel steel is its superior hardness. This can be attained in greater degree with a 20-per cent carbon-steel by case-hardening. This is being done by the Franklin and a few other companies. Some time ago it was the general practice to make the crankshaft bearings of six-cylinder engines about one-half the diameter of the cylinder-bore. This, with alloy-steel, was more than ample for strength, but the demand for a quiet engine has made it necessary to increase this figure greatly. A crankshaft-bearing 2 in. in diameter for a six-cylinder passenger-car engine having a 4-in. bore is not large enough today to avoid what is considered excessive torsional vibration and the consequent noise, and some designers are using a bearing 3 in. in diameter for an engine of this size. It would seem ridiculous to use chrome-nickel steel for a crankshaft approximately 3 in. in diameter, which apparently it must be to get rid of torsional vibration, when, with so large a diameter, plain carbon-steel will give a factor of safety that is greater than necessary. Often, in engineering development, we find it desirable to go back to something that has been abandoned. This applies also to the passenger-car underframe. If we must use a 9 to 10-in. channel to eliminate deflection, to get rid of body squeaks and the like, it is unnecessarily expensive to use chrome-nickel steel, because, with a frame of such depth, plain heat-treated carbon-steel will give as large a factor of safety as is necessary.

I think that the high-speed engine that Europe is adopting so extensively is not altogether a move in the right direction. This depends largely, of course, on the price of the car that is being built. I believe that a medium-speed engine is the best for a high-priced car, that the sensation of the excessive speed of rotation that can be felt in the high-speed engine will be condemned by the wealthy buyer, and that he will demand that his car be equipped with a medium or even slow-speed engine.

PROFESSOR UPTON:—The high-speed engine ultimately will not be the most popular. In Europe, engines are taxed according to the bore and, consequently, are being reduced in size and the speed is being increased. Some day, when the method of taxation is changed, the design of engines also will change. The speed at which lubrication is safe will limit the high-speed engine. Mechanical efficiency is higher in the medium-speed engine than in the high-speed engine. In the latter the inertia forces vary with the square of the speed and become large in comparison with the useful loads. The highest mechanical efficiency in both the high and the medium-speed engines will be found to be at about 1500 or 1600 r.p.m.

JOHN C. TALCOTT:—Professor Upton recommends the use of carbon-steel for frames. I believe that a frame can be built lighter with alloy-steel than with plain carbon-steel. It can be braced so that it will be stronger than a straight carbon-steel frame. I have seen straight carbon-steel frames break on trucks. If an alloy-steel frame gives trouble, it is better to add 10 lb. in braces than 60 lb. in the side-members.

PROFESSOR UPTON:—What I had in mind in recommending a return to carbon-steel frames was that the frames are being made deeper. We have not gone very far into this problem. Body weaving, however, is not due to the whole frame, but to one side being moved round. The reason is that we have not designed the whole thing so that it will be free from twisting. Twisting might not be objectionable in a truck frame, but on passenger cars it is objectionable. Daimler, in England, mounts the body in such a manner that it does not touch the frame, except at four points. It is fastened at the forward end by two hinges and at the rear by two rubber bumpers. This arrangement lets the body work on the frame. The forward end is flexible on the hinges, and they do not have to make the frame stiff; they make the body stiff and let it work on the frame.

MR. BURKHARDT:—Trusses usually localize stresses. They do not do anything else than brace up a weak point. In railroad cars, they can be of a certain design, but in the automobile the complications are greater and the best locations for them are rather doubtful.

(*Concluded on p.* 312)

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Single-Valve Internal-Combustion Engine Design and Operation

By R. Abell[1]

New England Section Paper *Illustrated with* Photographs and Drawings

SPECIFICATION is made of 13 essential factors that the author believes are necessary for attaining better engine performance; he then describes how improved results have been obtained by the use of a single-valve mechanism on an engine having an unusually high compression-ratio and using ordinary gasoline as fuel.

The latest type of engine built with the single-valve mechanism is illustrated, and detailed descriptions of its development, design and operation are given, together with comments upon its factors of advantage and data obtained from tests.

In designing this single-valve mechanism, the primary object was to solve the problem of positively closing poppet-valves without using springs and thus to produce a type of engine that would operate efficiently at greater speeds than are possible with spring-closed valves. Another object was to reduce the temperature of the exhaust-valve. The author claims that the design under consideration overcame disadvantages, such as carbon deposit, regrinding of the valves, overheating and detonation, and permits an increased compression-pressure and a consequent gain in power and efficiency.

IN the past few years a great amount of scientific investigation has been made on gas engines in the United States and in Europe. Problems have been solved that fill the gaps in engineering data to such an extent that we can discuss engines now on a more definite basis. Ricardo has given us some definite values of the increased efficiency obtained when using the higher compression-ratios that are permitted when additional exhaust gas is used in the fuel mixture. The Midgley indicator has set before us graphically the causes of detonation and has shown how some "doped" fuels help to eliminate detonation in an ordinary engine. During the last year, several very interesting laboratory tests have been run on engines operating with unusually high compression-pressure which showed great gains both in power and economy. Fuel-consumption values as low as those of Diesel engines have been obtained when using the ordinary four-stroke-cycle principle. One of these tests was run with 500-lb. per sq. in. compression-pressure with benzol as the fuel and resulted in an indicated mean-effective pressure of 176 lb. per sq. in. Another test made with ordinary fuel under a 120-lb. per sq. in. compression-pressure resulted in a 25-per cent increase of power over that obtained with its normal compression-pressure of about 80 lb. per sq. in. In the first case, results were obtained with a special fuel; in the second, they were attained by using special laboratory equipment by which the temperature of both the cooling-water and the intake gases could be controlled accurately. These tests are of value inasmuch as they set a high mark for engineers to shoot at, but appear to be of not more than academic interest to the automotive industry for the reason that neither method is commercially practicable. Recent investigations have shown that the key to the situation is the finding of practical means to reduce the temperature of the combustion-chamber surfaces, especially that of the exhaust-valve, which is the hottest part of the chamber in all engines.

The extreme heat of the exhaust-valve often has been stated to be one of the sources of detonation, but no definite value has been placed on it as a trouble maker because no method of segregating its influence on detonation from the other contributing elements has been known. Having built engines both with and without hot valves, I can state definitely that the relatively high temperature of the exhaust-valve is the chief cause of detonation.

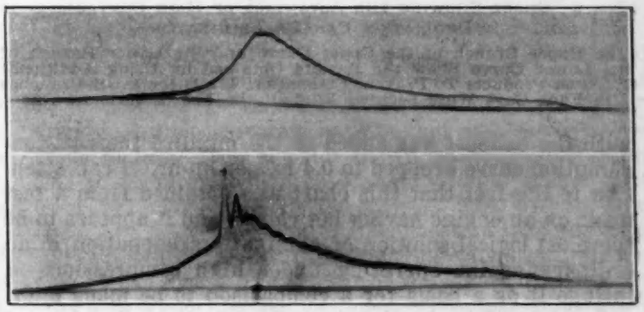


Fig. 1—Two Indicator-Cards, the Top One Showing a Normal Combustion and That Underneath One in Which Detonation Occurred

Fig. 1 shows two Midgley indicator-charts, the upper one being characteristic of an engine operating normally. While the pressure line shows very faint irregularities, it is substantially a smooth curve. The dark spots indicate the beginning of detonation. A slight variation will always be seen in the smooth lines before the knock is audible. The lower chart is a good illustration of the way a curve will vary with detonation. In this case the pressure drop is nearly 50 per cent of the maximum on the first fluctuation.

Fig. 2 shows a chart made by Harry R. Ricardo, of London, that is especially valuable since it shows the very great advantage of high compression, provided detonation can be eliminated. The upper branch of the upper curve shows the gain in power from high compression, using exhaust gas to prevent detonation. The lower branch of this curve is what occurred with the same engine under the same conditions without the addition of exhaust gas. Detonation is characteristic at a compression-ratio of about 4.7 to 1. The lower curves show the fuel consumption as applied to the two upper curves. At a compression-ratio of 7.5 to 1, without the exhaust gas, the fuel consumption came back to 5.5 lb. per hp-hr.;

[1] M.S.A.E.—General manager, Single Valve Engine Co., Mattapan, Mass.

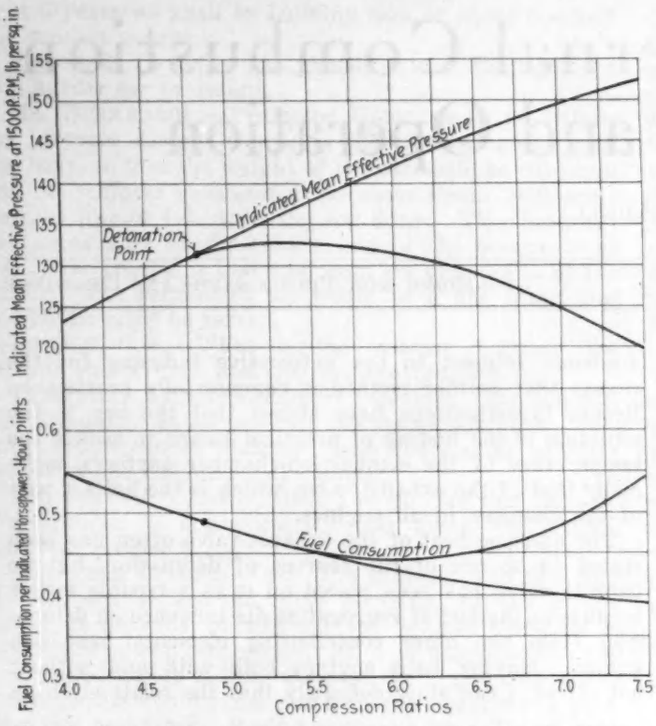


FIG. 2—CHART SHOWING THE ADVANTAGE OF HIGH COMPRESSION IF DETONATION CAN BE ELIMINATED

The Upper Branch of the Upper Curve and the Lower Branch of the Lower Curve Show the Results Obtained by Using Additional Exhaust-Products To Prevent Detonation, While the Inner Branches in Each Case Were Secured Without the Use of Exhaust Gas

with the exhaust gas added to the mixture the fuel-consumption curve dropped to 0.4 lb. per hp-hr. I call attention to the fact that this chart was obtained from a test made on an engine having hot valves, and it appears to be the most logical solution of eliminating detonation in an ordinary engine operating under high compression. I include it as a basis for a comparison to be made later.

So far as I know, the gas engine is the only piece of mechanism that operates with any of its vital parts at a red heat; and, irrespective of the improvements in valve materials that permit a fair amount of durability when operating at high temperatures, a gradual deterioration and a loss of power that call for expensive overhauling occur. This intense heat is the principal obstacle standing in the way of obtaining commercially the results that all engine designers are striving for, namely, higher efficiency and greater durability.

ESSENTIAL ENGINE-PERFORMANCE FACTORS

The essential factors necessary for better engine performance are:

(1) Higher compression
(2) A lower maximum temperature of the combustion-chamber and the valves
(3) A greater valve capacity
(4) Positive valve action at all engine speeds
(5) A compact combustion-chamber without pockets
(6) Means for vaporizing the fuel completely
(7) Uniform fuel distribution
(8) Adequate ignition
(9) A proper location of the spark-plugs
(10) Light reciprocating parts
(11) A rigid crankshaft
(12) Efficient cooling and lubricating systems
(13) Silent operation

The particular type of engine shown in Fig. 3 is one of four sizes that have been built to date, this being the last. In this case, the fan is mounted on the front end of the camshaft, but it was never used. The lower part of the engine, up to the head, was purchased from a firm in Newark, N. J., that has built some highly efficient engines having overhead shafts but with the standard two-valve arrangement. The part I supplied for this engine is from the cylinder block up.

Several important things are illustrated in Fig. 3. First, the aluminum manifold is of the simplest design, being merely a flaring elbow with a partition coming down about half-way around the elbow to prevent surging of the mixture to some degree. The exhaust-manifold is also of aluminum. The spark-plugs are set at an angle of 15 deg. The end of the screw-plug evident at the rear end is for closing the hole in which the rotary distributor operates. In demonstrating the effect of temperature of the distributor, we remove that plug when the switch is thrown off and pull it out by hand. It is warm, of course, but it can be held in the hands when wearing cotton gloves. The manifold is only half of a circle in cross-section, the other half of it being in the head. It is a plain milling operation to run across the head and mill off the surface to which the manifold is bolted.

On this engine the accessory drives are transverse at the front, and Fig. 3 shows also where the generator is mounted. The other side of the engine exactly balances the side illustrated in Fig. 3 with similar mountings for the water-pump and the ignition apparatus. The starting motor is in the bell housing; it has a standard S. A. E. mounting. Two buttons are provided to facilitate lifting the aluminum cover. The center one is a hand nut and is the only means of holding the cover down. Only about 15 sec. is required to lift the cover and no tools are needed.

Fig. 4 shows two sections of this engine. The crankshaft is made from a solid billet and has main bearings of 2½-in. diameter. This shaft is a remarkable example of good balance at any speed. Partly on account of the large diameter of the bearings and partly on account

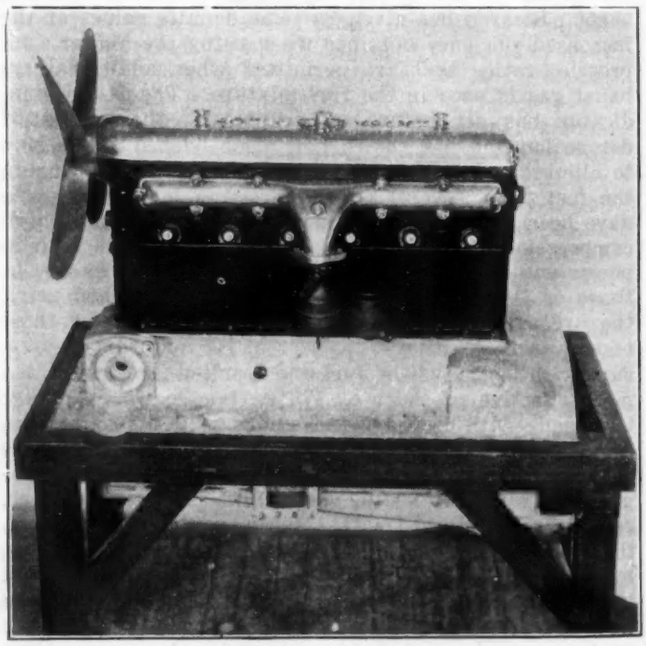


FIG. 3—EXTERIOR VIEW OF THE SINGLE-VALVE ENGINE, SHOWING THE SIMPLE DESIGN OF THE INTAKE-MANIFOLD AND CLEAN EXTERNAL APPEARANCE

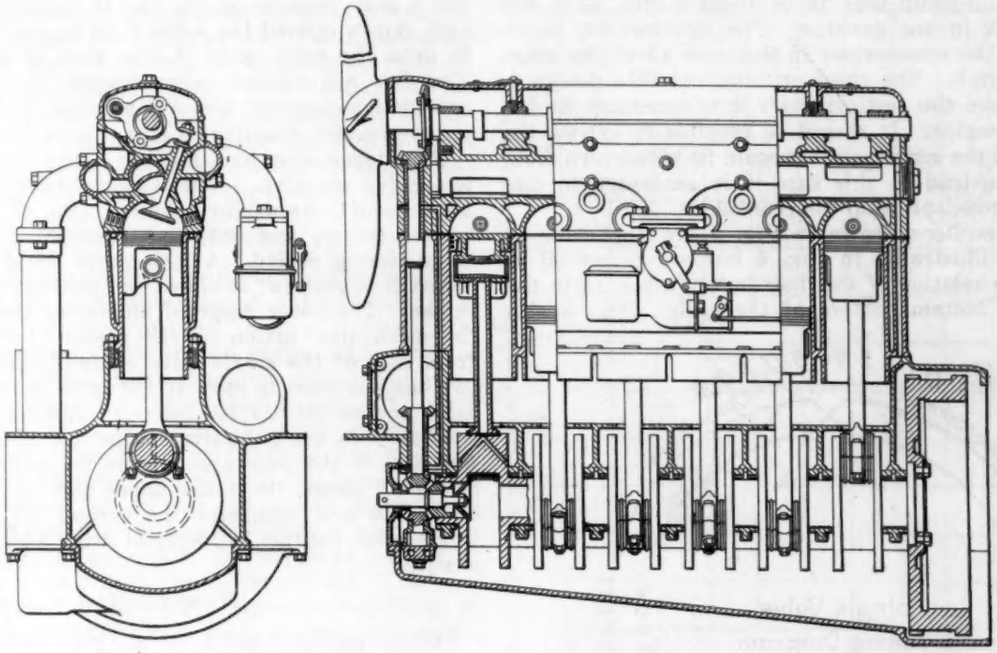


FIG. 4—END AND SIDE ELEVATIONS, PARTLY IN SECTION, SHOWING THE POSITIVELY OPERATED POPPET-VALVE AND THE ROTARY DISTRIBUTOR VALVE

of the light connecting-rods and the light pistons, the engine is practically free from vibration. The 10-in. rods weigh only 1½ lb. complete. The piston, with its rings and pins, weighs 1 lb. In this case, the Ricardo type of piston was used; it performed very satisfactorily and stood up well. The thickness of the piston-head is about 1¼ in.; the walls are about ⅛ in. thick below the ring grooves and 3/32 in. thick at the lower end. This is not a split type.

Regarding the construction of the cylinder block and the crankcase, the parting line comes at the top of the bell housing and a little higher than on the average job. The principal reason for this is to get the cylinder barrels to project into the crankcase to serve as oil traps. The main-bearing supports are webbed and form baffles over the crank discs. This prevents excessive oil being driven into the cylinder; such oil as does get by is trapped again in the case. The result is that, even though the oil is maintained at a high level so that the discs actually dip, it is impossible to make the engine smoke. This is due to the combination of the two baffles.

In regard to the head, the combustion-chamber is a true hemisphere. The valves are set on a slight angle to balance the design. This permits the camshaft to be maintained on the center line and also results in a more accessible valve mechanism that can be withdrawn from the head, from the side shown, without interference. The distributor, which is of very simple construction, is placed very close to the valve. The pocket which is formed by the distributor in connection with the valve, is regulated by its distance from the valve-seat. In this case the volume is 12 per cent of the piston displacement, which is 41 cu. in. per cylinder. The manifolds are in duplicate, one somewhat above the other. Except for a lack of accessibility, the vertical shaft-drive, which is mounted in bronze bushings at the top and at the bottom, was perfectly satisfactory. The oil-pump is bolted to the lower part of the crankcase and is of an ordinary gear type but it has no oil-pipe connection. It is bolted on to the bottom and the intake and the delivery ports are drilled in the crankcase and connected with steel tubes so that the pump can be taken off for inspection.

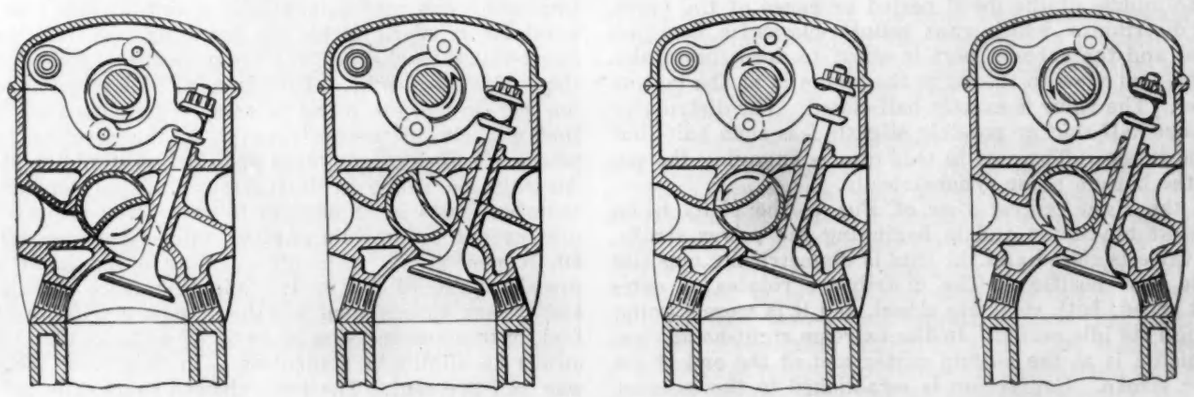
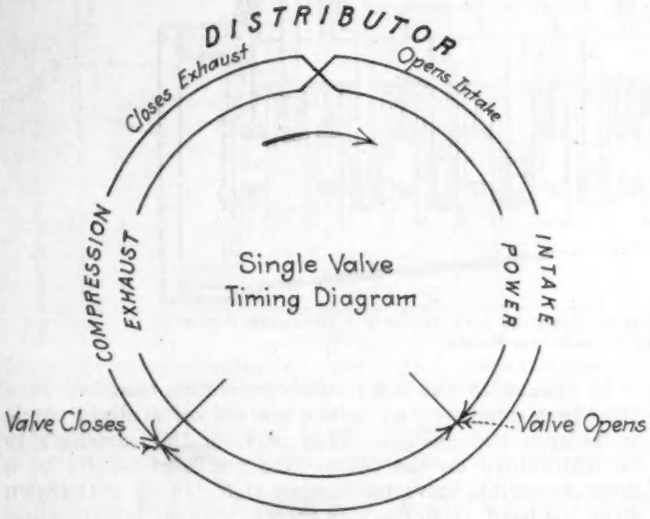


FIG. 5—FOUR SECTIONAL VIEWS SHOWING THE RELATIVE POSITIONS OF THE DISTRIBUTOR VALVE DURING THE CYCLE

From Left to Right They Show the Piston at the Top Center and the Exhaust-Valve Closed and the Intake Opening; the Piston at the Bottom Center with the Intake-Valve Closing; the Piston at the Top Center or the Beginning of the Power Stroke; and the Piston at the Bottom Center with the Exhaust-Valve Open

To put the oil-pump back takes about 2 min. as it will only go back in one position. The intermediate bevel-gear drives the accessories; in this case all of the gears are of 10 pitch. The chief criticism of this design is that to remove the vertical shaft it is necessary to dismantle the engine. It should be possible to expose the drive so that the gear assembly could be withdrawn from the front; instead in this case it is necessary to disconnect the rods and drop the crankshaft.

The four sections shown in Fig. 5 are duplicates of the sections illustrated in Fig. 4 but are presented to illustrate the relation of the distributor in regard to the top and the bottom centers of the cycle. Fig. 5 also



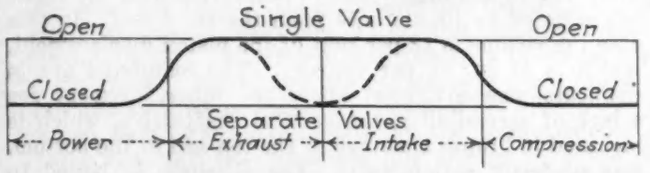


FIG. 6—TIMING DIAGRAM OF THE SINGLE-VALVE ENGINE (ABOVE) AND (BELOW) THE RELATIVE ACTION OF SINGLE AND DOUBLE VALVES

shows the relative position of the valve. In the extreme left-hand view the piston is at the top center; the valve is wide-open and the cam is in contact with the roller in the middle of the dwell period or pause of the valve. The distributor which runs counter-clockwise has just closed and the exhaust port is about to open the intake. In the next view to the right the piston is at the bottom center. The valve is exactly half-closed. The distributor is about half-shut or possibly slightly less than half-shut to the intake. The cam in this case is engaging the top and the bottom roller symmetrically.

In the right central view of Fig. 5, the pistol is on the next top-center and is beginning the power stroke. The valve is closed and the cam is diametrically opposite to its first position. The distributor rotates at camshaft speed; both ports are closed, and it is merely going through its idle period. In the extreme right-hand view the piston is at the bottom center and at the end of its power stroke. Connection is established to the exhaust port. The cam is opposite to the position assumed in the view to the left of the center. It should be noticed that the valve is comparatively large in diameter and has a port opening of 1¾ in.; it could be larger. The cam shown opened the valve 7/16 in.; it could be made to open the valve more if this were of any advantage. The two spark-plugs point toward the center of the combustion-chamber and are as close to the bottom of the combustion-chamber as it is practicable to have them.

The upper diagram in Fig. 6 has reference to the top-center position. The exhaust-valve opens at the usual period, the average timing being 45 deg. from the bottom center, and continues open until the usual exhaust-closing period. At this point the distributor performs its function; it closes the exhaust and opens the intake. The lower diagram illustrates the difference in the mechanical action of the present two-valve engine from that of the single-valve engine. Assume that the two engines start to open at the same time. The single-valve engine attains its full-open position at the same time as does the two-valve engine and it remains open; whereas, in the two-valve engine the exhaust-valve immediately closes, its intake opens and joins in with the other line and completes the cycle. The cycle of the single-valve engine is identical with that of standard practice.

OPERATION

The mechanism consists of one poppet-valve per cylinder and each valve is opened at the usual exhaust-valve opening-point; it remains open throughout the exhaust and intake periods and is positively closed at the usual intake closing-point. Each valve is supplemented by a rotary distributor, running at camshaft speed, which controls the flow of intake and exhaust gases to and from their respective manifolds.

To solve the problem of positively closing poppet-valves without using springs, and thus to produce a type of engine that would operate efficiently at greater speeds than are possible with spring-closed valves, was the primary object of designing this mechanism. Another object was to reduce the temperature of the exhaust-valve. The problem was solved, but several other advantages were discovered during the course of development. These advantages overcame many of the difficulties and the limitations that the engine designers and the public have had to contend with, such as carbon deposits, regrinding of the valves, overheating and knocking. At the same time, the design permits an increased compression and consequent gain in power and efficiency.

In the ordinary poppet-valve engine separate valves are provided for the intake and the exhaust gases. The intake-valve is maintained at a comparatively low temperature at all times by the incoming gas, but the exhaust-valve reaches a maximum temperature of more than 1200 deg. fahr. This is a red heat and is much too hot to permit using a compression-pressure value that will obtain the best results in the development of power, the fuel consumption and durability of the valves. An extreme range of temperature of the combustion-chamber exists in all engines to date. The compression-pressure of automobile engines ranges between 60 and 80 lb. per sq. in. gage pressure, or a maximum compression-ratio of 4.5 to 1. Scientific tests by Ricardo and others have shown a gain in power with ordinary fuel up to a compression ratio of 7.5 to 1, by using some means to eliminate detonation. In this case the gain was 18.7 per cent. The tests showed practically no connection between detonation and the temperature of compression.

Incandescent spark-plug points are often a contribut-

ing cause of detonation in combination with hot valves, but this difficulty has not been found in the development of single-valve engines having a compression ratio of 6.5 to 1, which has been employed with perfect success in six-cylinder engines used in a standard car on the road. It has been impossible to make this engine knock during 8000 miles of road test, even when the engine was stalled by using the brakes with the throttle wide-open.

Development History

Fig. 7 illustrates the first application of the single-valve mechanism to a water-cooled automobile-engine. A single-valve head was adapted to fit a Chevrolet engine; all of the lower parts of the engine were retained in their original form. Some difficulties were experienced in providing a drive for the overhead camshaft, but a compromise was made by using the water-pump shaft and running two drive-chains up overhead. In this case the valves were mounted vertically in the center of the combustion-chamber, which was not exactly hemispherical; the corners of the combustion-chamber were at an angle of 45 deg. The reason the combustion-chamber was not made to be a true hemisphere is that I did not appreciate the real value of the hemispherical form at that time. The distributor is made in two sections in this case, with a tongue connection merely as a convenience for the foundry in making the core which would be apt to sag if made in one piece and would be troublesome to handle. Helical spur-gears drive the distributor from the camshaft in this case.

When the dotted power-curve in Fig. 8 was taken, it was with an engine that was accurately fitted and that had been limbered-up; it was the best curve obtainable up to that time from that engine. When the new head was made, it was fitted to a new engine and was set-up so tight that when we put the engine on the dynamometer to determine the friction load, we found that 7 hp. was required to jack it over at 1000 r.p.m. With the former engine only 3 hp. was necessary to jack it over at the same speed; so, there was a handicap of 4 hp. at 1000 r.p.m.

In the application of this single-valve principle to the Chevrolet engine, only standard parts were used with

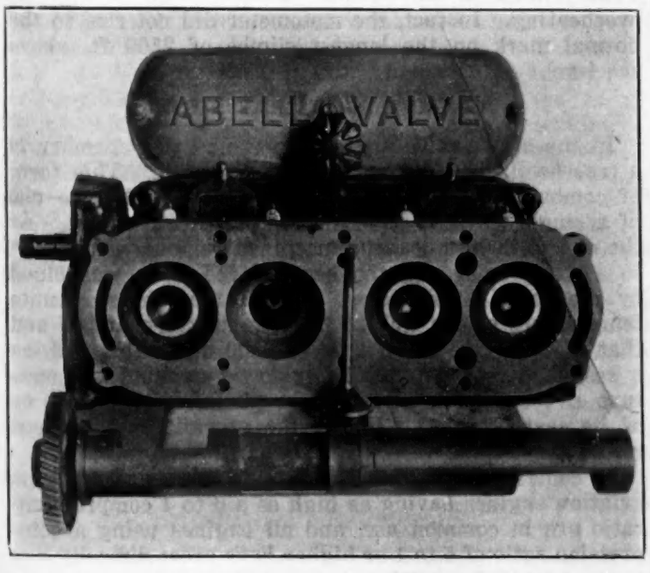


Fig. 7—The First Application of the Single-Valve Mechanism to a Water-Cooled Automobile-Engine

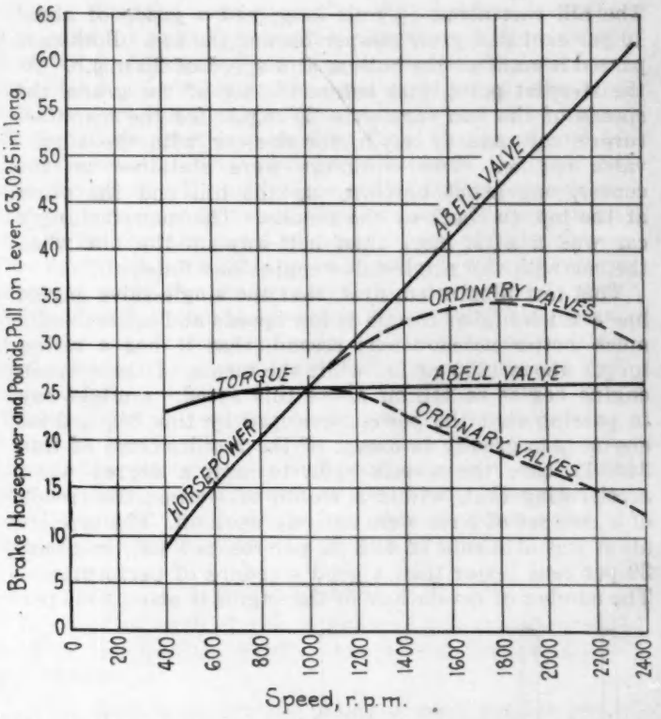


Fig. 8—Results of a Dynamometer Test of the Single-Valve and the Conventional Cylinder-Head

the exception of the cylinder-head and the valve mechanism. The compression pressure used on this engine was about 115 lb. per sq. in. gage. At a speed of 2000 r.p.m., the engine with the two-valve head developed 34 hp., but with the single-valve head it developed 51 hp., an increase of 50 per cent. At a speed of 2400 r.p.m., the original engine dropped to 30 hp. but, with the single-valve mechanism, it climbed to 62 hp. This was the highest speed at which the dynamometer tests were made, owing to severe crankshaft vibration resulting from using a crankshaft that was too small in diameter to carry the increased power safely. The power curve with the single-valve mechanism was a straight line and showed a slightly increasing torque.

An application of the single-valve mechanism was then made to a 3½ x 5-in. six-cylinder Trego engine of 288 cu. in. displacement. The first test of the engine at 2000 r.p.m. on a dynamometer showed 64 hp. with a compression pressure of 80 lb. per sq. in. The construction of the engine to which the head was adapted would not permit the increasing of the compression pressure without making a considerable change, but we did gain a decided increase in power over the best obtained with the original valves.

The fourth engine equipped is shown in Fig. 3. This is a 3¼ x 5-in. engine of 249-cu. in. displacement having a 115-lb. per sq. in. gage compression-pressure. This engine was mounted in a standard Roamer car and given road tests of 8000 miles, throughout which it showed unusual performance. Soon after this car was put into commission, a hill-climbing test was made by a car builder, who tested this engine against his standard make of six-cylinder engine in a car that was showing good average road-performance. A comparison was made between the two cars, taking into consideration the size of the engines, the weight of the cars, the size of the wheels and the rear-axle gear-ratios. These figures gave the manufacturer's car an advantage of 12 per cent.

The hill was about ½ mile long, had a grade of about 10 per cent and grew steeper toward the top. Both cars passed a mark at the bottom at a speed of 20 m.p.h. At the steepest point, just before the top of the grade, the speeds of the two cars were 20 m.p.h. for the manufacturer's car, and 41 m.p.h. for the car with the single-valve engine. Two observers were stationed on the course, one about half-way up the hill and the other at the top, to check-up the results. The manufacturer's car was a little more than half-way up the hill when the car with the single-valve engine had finished.

This test illustrated, first, that the single-valve engine has a much higher torque at low speeds and consequently much better acceleration; second, that it has a rising torque above 1200 r.p.m., while the torque of the average engine begins to fall off above this speed. I might say in passing that the power developed by this 3¼ x 5-in. engine was greatly in excess of the requirements of this 3400-lb. car; the wheels spun to such a degree when accelerating that, within a month of testing, the treads of a new set of tires were entirely worn-off. The acceleration was at a rate of 4.18 ft. per sec. per sec., or about 30 per cent better than a good standard of performance. The number of revolutions of the engine is about 2500 per

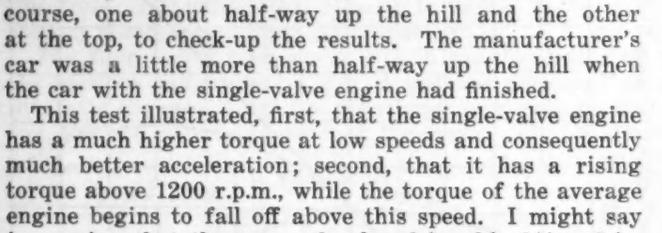


FIG. 9—THREE TYPES OF COMBUSTION-CHAMBER

That at the Extreme Left Is a Typical Two-Valve-in-Head Engine in Which the Corners Form Pockets; the Central Chamber Is a Single-Valve Engine with the Corners Beveled and That at the Right Is a True Hemispherical Combustion-Chamber with No Corners or Pockets

mile, which is only a moderate speed for this engine. In other words, a much smaller engine geared to turn about 3000 revolutions per mile would give a better performance than the average on acceleration, at least 70 m.p.h. at top speed, and a greater fuel- (per gal.) mileage.

FUEL CONSUMPTION

Regarding fuel consumption, the test car, which weighed with a passenger about 4000 lb., showed about 25 miles per gal., or approximately 50 ton-miles per gal. of gasoline, which is above the average. On a fuel test at speed, on one of the long concrete roads in Michigan where a maximum speed of more than 70 m.p.h. was reached several times, the fuel consumption was 21 miles per gal. No outside heating arrangement was used for the intake, and little need for it was found. This is accounted for by the fact that the unvaporized particles of fuel in the intake-manifold are warmed-up by contact with the distributor and the poppet-valve which, in turn, is thereby maintained at a practically constant temperature of 500 deg. fahr. The incoming gas is also warmed by the absorption of the small amount of exhaust gas that is left in the port between the distributor and the valve, and this is a decided help in completing vaporization. Moderate heat applied directly above the carbureter would increase the fuel mileage in cold weather.

Ricardo states that, for every 1 per cent of exhaust gas added, the *flame temperature* is lowered 45 deg. fahr. Thus, with the 12 per cent in the distributor and in the pocket, the flame temperature is lowered to a considerable degree. All this, coupled with the fact that the hemispherical combustion-chamber is conducive to more complete combustion than is possible with other forms, results in less heat being reqired in the manifold. Another way to explain this function is to say that it is the incoming gas that prevents the excessive heating of the poppet-valve, and that this excess heat, being transferred to the incoming charge, does useful work and results in a local hot-spot for each cylinder at the most effective point, accomplishing the desired result without the usual hot-spot. This heat transfer is especially valuable for the cold engine, because some of the heat from the exhaust of the very first explosion is picked up by the incoming charge and results in normal operation in a very few seconds even in cold weather. It is not necessary to enrich the mixture until the entire engine has become heated to its normal point. The effect is similar to that of the fuelizer and is accomplished automatically without using an extra volume of fuel but is the product of factors inherent with the single-valve design and so requires no additional mechanism. Popping back through the carbureter is practically eliminated.

SEPARATE-VALVE DIFFICULTIES

Some of the difficulties met with in an engine having separate valves for the intake and for the exhaust are: first, that the temperature surrounding the exhaust-valve seat is so high that water adjacent to these surfaces boils and causes an increased temperature of the cooling water. This, in turn, requires a larger radiator and a more efficient cooling system than is needed in the single-valve type; second, when two or more ports enter the combustion-chamber, they occupy space that otherwise would be available for cooling surface. With the single-valve, only one port enters the combustion-chamber and there is not only an entire absence of boiling water near the valve seat, but about 25 per cent more of the outer surface of the combustion-chamber is available for cooling. With an undersized radiator and no fan, the test car was driven through the mountains of western Pennsylvania in hot summer weather with no sign of overheating. In fact, the motometer did not rise to the normal mark on the longest climbs of 2500 ft. above sea level.

COMBUSTION-CHAMBER FORM

In the single-valve engine the combustion-chamber is a true hemisphere with a machined surface. This form of combustion-chamber, which is permitted by the use of a single valve, is a decided factor of advantage. At the end of 8000 miles, the machined surface was turned to a straw color and the head of the valve was blued by the heat. These colors indicate that the maximum temperature of the wall of the combustion-chamber and that of the valve did not exceed 500 deg. fahr. and, as a result of this comparatively low maximum temperature of the valve, no warping or pitting of the valve or of its seat occurred. Hence, the necessity of frequent regrinding of the valves is eliminated.

In contrast to this, it is interesting to note that no aviation engines having as high as a 6 to 1 compression-ratio are in common use, and all engines using a compression ratio of 5 to 1 or higher have great difficulty due to the warping or pitting of the exhaust-valves and their seats even when special alloy-steels are used. This

necessitates frequent replacements and labor or expense to keep them running satisfactorily. Of course, the use of higher compression would result in a greater amount of horsepower if it were not for the fact that the charge would detonate badly because of the hot exhaust-valve. When the valve is kept below the temperature that causes warping and pitting, it is no longer a contributing cause of detonation.

In Fig. 9 the diagrams reproduced further illustrate the points I have mentioned in relation to combustion-chambers. At the left is a typical diagram of a two-valve engine. The valve is in the head and is obviously at a disadvantage with the other diagrams so far as carbon deposits are concerned, because the gases follow different paths in entering and in leaving the cylinder and the walls of the combustion-chamber are not being swept evenly by those gases. While the gases will follow a streamline, the corners form dead pockets; the reverse would be true when the other valve functioned. I believe that the middle diagram of Fig. 9 illustrates a better combustion-chamber than any now being used, but this one was defective, as we found it. After two weeks of running on the dynamometer the head was removed and we found carbon deposited in the corners where the trouble had started. The true hemispherical form of combustion chamber is shown in the diagram at the right in Fig. 9. When gas enters from the valve, it must of necessity sweep those walls uniformly and, on leaving the cylinder during the exhaust stroke, it must sweep them again. So, the flow of gas has a great influence in keeping the oil-film in constant motion and does not permit it to come to rest for a long enough period to burn-on any carbon.

For the first time the practical use of the hemispherical combustion-chamber is available in combination with the single-valve arrangement, because the port area is not limited, as is the case with separate valves. The single valve can be three-fifths the diameter of the cylinder, which is a greater area than it is possible to obtain even with four valves per cylinder as used on some aviation and racing-car engines. Not only can a larger single-valve be used in a given-sized cylinder, but it can be opened farther because its closing is positive instead of being dependent on the limited speed of springs.

To finish the hemispherical combustion-chamber in one operation is a very simple machining operation, and it has been discovered that this type of chamber, in connection with the single valve, prevents the accumulation of carbon deposits even though the engine is run with an over-rich mixture or with a piston that pumps oil. The reason for this is found to be that the gases passing in and out of the combustion-chamber through the same path keep the oil film in constant motion and, as no pockets are provided for the oil to lodge in out of the streamline of the gases, it cannot come to rest for a long enough period to carbonize, especially when the temperature of these walls is maintained at such a comparatively low point. These features make it evident that the outstanding advantage of a single-valve engine is a thermal one and that the lower temperature of the combustion-chamber walls eliminates the difficulties encountered in the past and permits the use of much higher compressions.

Mechanical Features

In considering the mechanical features of this design, I refer first to the distributor, which is a single revolving element made of cast iron and cored-out to save weight. For the average engine the diameter is about 2 in., and an ordinary running clearance of from 0.003 to 0.005 in. is used. From the beginning absolutely no difficulty due to sticking has been experienced; the distributor does not get hot enough to distort or to burn-off the oil-film that is picked up from the gases and is always present. The distributor controls the flow of both the hot and the cold gases and a moderate temperature is consequently maintained, never becoming high enough to color the metal even to a shade of light straw. The distributor should not be confused with a rotary valve, as such, for the very obvious reason that it does not hold either the compression or the explosion pressure nor does it come into contact with the gas during combustion. The poppet-valve holds the compression and the explosion pressures and thereby shields the distributor from any pressure except that in the manifolds, which is comparatively low. For this reason, the oil-film on the distributor is sufficient to seal it effectively and the engine idles as well as any other. The test car could be throttled down to less than 3 m.p.h., and it accelerated rapidly without any sign of loading.

Manifold Design

One-half of each of the intake and the exhaust-manifolds is cast into the head. This is done for the following reasons:

(1) Such a construction makes a much simpler job for the foundry, because the cores are supported the entire length of the head and act as gas vents

(2) The casting can be inspected better for imperfections, since it is possible to look straight through the ports from one side to the other

(3) If desired, the intake-manifold can be polished on the inside

(4) A very compact arrangement is secured; the manifolds project less than half the usual distance

(5) One-half of the intake-manifold is heated by the water-jacket to a desirable temperature and, on the other hand, one-half of the exhaust-manifold is cooled to a desired temperature so that the difference in the amount of heat escaping under the hood is considerable

(6) About 30 per cent less water-cooling system is required than with other engines of equal size

(7) Aluminum exhaust-manifolds can be used without danger of melting

The sectional view shown in Fig. 10 is included for reference with regard to manifold and other design features. I focused my attention on the positive operation of the valve and drew charts to determine what kind of cam would be required to eliminate lost motion.

The fulcrum point for the rocker-arm is integral with the bearing cap which must, of necessity, fit the camshaft. In this case the rollers have a 3/4-in. outside diameter and each has a 7/16-in. hole. They revolve on hardened bushings having 5/16-in. holes. An ordinary cap-screw is used to hold the bearing cap in place. If, after a long enough time, the distance between the rollers is enough to cause noise, we simply pull the bolt out, remove the top rollers and replace them with others that are over-size. The connection to the valve-stem is merely a rounded end; it would be made by a stamping process and ground to size. The collar is an integral part with a castellated head and is a simple machining job. It has been suggested that this collar be

made in two parts or made adjustable so that the wear can be taken up. I have made such a collar, but it is not practicable because a compound adjustment is required on a small part that is not worthwhile. If it wears out, a few cents will buy a new one; so, it is not worthwhile to make it complicated.

Positive Valve-Control Mechanism

Of all the mechanical features of this engine the most interesting is the mechanism for positively controlling the action of the poppet-valve. This valve opens with exhaust-opening and closes with intake-closing timing so that the cam shape is very different from that used with separate valves. The cam operates a rocker-arm having two opposed rollers, one of which acts for opening and the other for closing the valve. For proper timing of the valve, the cam is of such a form that it is in contact with both rollers in all its positions.

The rocker-arm is of such a length that its angular motion is relatively small and the valve opens twice the distance of the cam-action throw. With a valve opening 7/16 in., the lever moves only 3 deg. on either side of its central position and produces the minimum side-thrust on the valve-stem. Thus, the simplest possible mechanism fills the requirements.

The cam shown in Fig. 10 opens the valve 45 deg. before the bottom center and stays open; the distributor shifts the manifolds at the top center and the cam closes the valve at 45 deg. after the bottom center on the compression stroke. Thus, the valve opens but once for each power impulse.

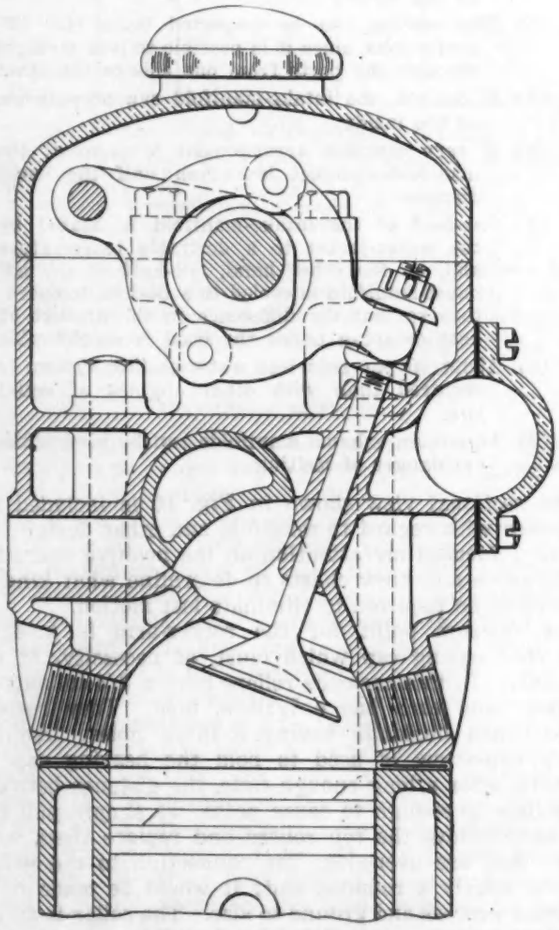


Fig. 10—Partial Section of the Cylinder-Head of a Single-Valve Engine

Another important detail is the difference in the rate of acceleration and deceleration of the valve in opening and closing. The valve leaves the seat at about four times the speed with which it approaches the latter in closing. This is desirable for power and for silent operation. The variation in the speed of the valve is due to the relation of the angular motion of the rocker-arm to the direction of rotation of the camshaft. This action corresponds to that of a cam having a carefully designed ramp but, in this design, it is accomplished by the combination of the arrangement of the rollers on the rocker-arm and the direction of rotation of the camshaft. The valve action is unusually silent and the valves do not chatter at full speed as is the case with a spring-closed valve.

Valve Adjustment and Gearing

The adjustment of the valve is by the grooved collar threaded to the valve-stem and provided with a castellated head for a cotter-pin. In adjusting the valve the proper procedure is to draw the valve to the seat, back it off one notch and then insert the cotter-pin. It is not necessary to hold the valve against its seat mechanically. About one-half the usual amount of tappet clearance is used. This system cannot be used with separate valves, as the exhaust-valve would be drawn from its seat during the suction stroke. The valve is constantly turning in relation to the valve-seat, since there is no spring to restrict it.

This type of engine requires a vertical-shaft drive for the overhead camshaft, and I have found it satisfactory in every way. The usual gear-noise is not present, because there is no reversal of torque in the shaft due to the absence of valve-springs and also to the fact that there is but one-half the number of valves to operate, which reduces the load on the shaft to less than one-half that which is present in other engines.

Two advantages of using gears for the overhead drive are (a) that they are about one-half the diameter required in the present standard types and so the tooth speed is correspondingly less and (b) that they can be adjusted accurately for meshing properly, which is not possible with gears on parallel shafts that have fixed centers. The cost of the four small gears does not exceed that of the usual two large ones. It is universal practice to use the overhead camshaft on aviation and racing-car engines; many of the later European passenger-cars and a few cars that are built in the United States, are using it successfully, also. I predict that the overhead camshaft will become the standard method in the future because of the increased efficiency and accessibility that it provides.

One very great advantage of the overhead-camshaft design is the extreme accessibility. Because the entire valve mechanism is uppermost, close adjustment can be made with little effort and no allowance need be made in adjusting the tappet clearance for a hot and a cold engine as is required with long valve-stems and push-rods. The entire head can be removed within 15 min., and the entire valve mechanism can be adjusted on the bench if desired. For a test, I have removed the camshaft from the engine in 4 1/2 min., purposely disarranging the adjustment of the valves while doing so. The reassembling of the camshaft and the adjustment of the valves was accomplished within 6 min. This reassembling time cannot be approached with any other type of engine. The timing of the camshaft and of the distributor is done after the

engine is assembled and entirely by gears in the head. A vernier adjustment permits a setting within 2 deg.

Cost

Contrary to the popular idea, a well designed overhead-camshaft engine is no more expensive to build than other types. Owing to the comparative simplicity and compactness of the crankcase and the cylinder block, a considerable saving is effected in the cost of these parts and in the weight of the engine. This 3¼ x 5-in. six-cylinder engine without its accessories weighed only 360 lb.; the small weight is accounted for partly by the fact that the greatest width of the cylinder block or head is 5 1/2 in. Also, the machining operations cost less because the design of the mechanism is simpler than is usual. The head for an overhead-camshaft engine is more expensive to build than that for other types but, as the mechanism is more compact, the cost of the parts is less and the total saving in weight for the complete engine is about 20 per cent. Both material and labor costs are less than for an L-head engine of an equal bore and stroke.

THE DISCUSSION

V. A. NIELSON:—How is the distributor lubricated?

R. ABELL:—The engine that was built in Detroit had no means provided for lubricating the distributor other than such oil as it got from the engine cylinder. When the engine was put on the dynamometer for test, I put the distributor into its bearings when it was bone-dry; an oil-film was established automatically as soon as the engine started.

The Trego engine was built with a distributor 30 in. long, in one piece, and no trouble due to distributor warpage occurred. A means of lubricating the distributor was provided by drilling small holes from the chamber containing the camshaft in between the ports, so that the oil which always is present in the head would trickle down through these holes and provide sufficient lubrication; but, within about 30 min., we plugged these holes shut because they provided oil enough to make the engine smoke. The distributor never required any additional lubrication. Some oil is always going out with the exhaust gas, sufficient in amount for this purpose.

QUESTION:—How much clearance is allowed between the distributor and its bearing?

MR. ABELL:—The distributor clearance is about the same as is allowed for the piston. It appears not to make much difference whether the clearance is 0.003 or 0.006 in. I have run a distributor that had a clearance of 0.012 in. and it did not show 1 lb. of difference on the dynamometer. When the engine is idling at low speed the amount of clearance makes a slight difference. This results in a greater amount of dilution; that is, the vacuum in the intake draws over more exhaust-gas and dilution occurs to such a degree as may cause a cylinder to miss and thus develop irregular running; but the engine will not stop. With a large distributor clearance, the engine will run at less than 180 r.p.m.

QUESTION:—Why does not the cooling water heat-up under the conditions you have specified?

MR. ABELL:—The explanation is complicated; the four contributing factors are

(1) The valve does not reach half the present maximum temperature of the ordinary valve, and no hot-spots are present to boil the water

(2) The hemispherical combustion-chamber presents a minimum of heat-absorbing surface to the hot gases and has a maximum of cooling surface in contact with the water; in addition, it permits more complete combustion

(3) The higher compression-pressure permitted also improves the combustion so that there is a lower average temperature of the exhaust gas

(4) The additional exhaust-gas taken back with the new charge reduces the maximum flame-temperature by several hundred degrees fahrenheit.

All of the four factors enumerated, considered together, reduce the temperature of the cooling water as compared with that of ordinary types of engine having separate valves.

Ricardo states that, when changing compression on his test engine, the exhaust-pipe will become red-hot for a certain distance under the lower compression-pressure. He says that when he increases the compression-pressure, the red-hot section is shortened in proportion to the amount of the increase in the compression-pressure. He says also that each 1 per cent of exhaust gas added to the fuel charge reduces the flame-temperature 45 deg. fahr. This reduction in the flame-temperature was what I sought to accomplish.

QUESTION:—Have you made any test in which you used only one spark-plug?

MR. ABELL:—All tests were made when using single ignition.

QUESTION:—What is the amount of pressure of the valve against its seat?

MR. ABELL:—The only pressure against the valve-seat is that produced by inertia and by gas pressure. The valve is not pulled against its seat mechanically. When the engine is not running, the valve is not resting against its seat; by taking hold of the valve-stem, the valve can be spun around. The adjustment is to draw the valve against its seat, so that it can be felt to touch its seat, and then to back it off for 1 or 2 points of the castellated nut and insert the cotter-pin.

QUESTION:—Have you ever operated the engine with a combustion-chamber that was not machined?

MR. ABELL:—Yes. On the Chevrolet engine used, the combustion-chamber was not a true hemisphere and it was not machined. Running under 115-lb. per sq. in. gage pressure and after a two-week run, slight signs of "pinking" were noticed. We found some traces of carbon deposit in the corners of the combustion-chamber, but this would not have occurred if the combustion-chamber had been a true hemisphere in form.

Engineers and Production Managers

By Charles S. Dahlquist[1]

Cleveland Section Paper

After generalizing on the benefits conferred on industry and thus transferred to mankind by the engineer, the author defines the duties of the engineer and of the production manager and describes the attitude of mind that each usually has toward the other, an attitude that is too often unfavorable.

Asking whether this is due simply to a lack of cooperation, to the differences in their training and education or to the differences in the requirements of the two positions, the author analyzes these requirements in some detail and concludes that the engineer, as the creator, should have the confidence and the enthusiasm of an organization, supply all of the departments with the energizing power and be the dynamo; and that the production manager, as the producer, should embody the conservatism of an organization, be the motor and not the dynamo. The two positions should constitute a close working harmonious unit.

THROUGHOUT the last century, the engineer has given marvelous evidence of a continuous growth. His development of towns, cities and countries exemplify this; as do also his development of industry, his employment of labor and the improvement of labor conditions, the phenomenal advance in numbers of products, his development of the modern home and his service to the entire world of intellectual people and barbarians as well. Examples of some of the late proofs of his service are the automobile, the airplane, wireless communication and electrified railways. Thousands of other improvements are products of the brain of the engineer; he is the creator and is recognized as such by the populace. The names of Edison, Steinmetz, Marconi, Haynes, Ford and many others are all household words. They are engineers in the progress of science, of invention and of service to humanity; they have given proof of the growth of engineering service. The engineer stands preeminent in the shouldering of responsibility and in the safeguarding of human life. The safety of a bridge, of a building, of riding in an automobile, a train or any passenger conveyance is due to the training, perseverance and knowledge of the engineer. His work is not alone the subject of inventive planning and building; he must always have in mind that a deep and far-reaching responsibility is his, because humanity suffers for any false move or wrong calculation. In no other profession are as exacting demands made.

When we stop to consider the possible consequences of the mistaken judgment of an engineer, we can realize his responsibilities. We, as engineers, invent, plan and build, having before us at all times safety, service and satisfaction. We must build well, serve well and please the critical mind of a judging public. That the engineer has accomplished these results is fully demonstrated in our everyday lives as we come into contact with the products of his endeavors. But he is not infallible, nor is he free from the criticism to which all men of initiative are subjected. He is an essential in the scheme of life, but only as cooperator with his fellows. Probably the nearest cooperator with whom he comes into contact is the production manager or the brain that makes the engineer's ideals a reality.

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Engineering and Production

Leaving out of consideration the subjects of finance, equipment and organization, although they are essential in the progress of engineering, the object of this paper is to point out the cooperative spirit that is so necessary between engineers and the men responsible for production.

When we pause to consider the position of the production manager, we must realize that his pathway is not one on which he goes Rolls-Roycing over boulevards. He has his sharp turns to make and his rough roads to cover; the semaphore is probably thrown to "stop" for him as often as it is for the engineer. However, the rough roads, sharp turns and stop signs do not bother a production man; he has the reputation of being a good driver. "Ask the men he has driven."

Many compliments have been paid the engineer by engineers, but let us put the engineer into the heart and the mind of the production manager and see ourselves as he sees us. Then our beautiful pedestal has crumbled away and, somehow, we have tumbled into the big mass of humans.

I have asked the production manager his definite opinion of an engineer. He thinks an engineer is a "nut" who sits in an office that has its walls plastered with diplomas, degrees and dust; who wears a woe-begone expression on his face, feeds on the end of a pencil and carries his brains on a slide rule; whose talk is confined to stresses, bending moments, section moduli, theorems and isms; who can tell you more about the thing you know nothing about and do not want to know and less about the thing he knows and thinks you should know, than any other being on earth.

But the production or the factory manager does not have a monopoly on thought. The engineer defines the production manager as a well-fed, ruddy-faced individual who sits in an office far removed from the engineering office and who thinks only in terms of jigs, tools, fixtures, machinery, cores and patterns and production figures. The walls of his office are usually a maze of charts, blackboards and "the schedule"; his favorite opening speech to those who enter is "Well," and his password to those who leave is: "Sure, we can make anything, but remember that this plant is not run as a charitable institution."

What is the reason for these two so opposite definitions? Is it due to a lack of cooperation on the part of engineers and production managers, to the differences in their training and education or to the differences in the requirements of the two positions in an organization? It may be due to a feeling on the part of both that they have not taken full advantage of their opportunities. The engineer may feel that he has not labored in the shops and familiarized himself with the problems of production. The production manager may realize that he has not fortified himself with a knowledge of engineering. Thus both men would be in a position of being always on their guard.

It is a perfectly natural desire on the part of each one of us to wish to be intelligent and to hold our own in discussion incident to our special business. As a rule, the production manager is a man who has worked his way

up step by step through the ranks at the factory; he has lived with the purr of the wheels in his ears and has rubbed shoulders with his fellow men from the sweeper to the superintendent; blueprints have been his Bible. He has learned that press fits are possible only in certain limit dimensions, that a straight line is the shortest distance between two points and that simplicity, accessibility and machinability spell speed, reduced costs and profits. His mind is trained to read a blueprint, not as a work of art, but as a dollars-and-cents proposition. His viewpoint of a blueprint is not as a finished product, but as a series of operations to be performed by machinery for which his factory is or is not fitted. He sees the doughnut from the mixing of the dough through its various stages to the final fried cake and its final conversion into "dough"; but the engineer sees a hole surrounded by a ring of browned, crispy, delicious cake that pleases the eye and appeases the appetite. Both men are optimists, but one views the doughnut as a commercialist and the other as an appeal.

However, it should be borne in mind that the appeal must always be made just before the instinct of commercialism has its opportunity and, in making his appeal, the engineer of today will discover that success requires of him a more complex knowledge of business essentials than was required of his brother of yesterday. That he confines himself to the problems of design and to engineering alone is not sufficient; he should fortify himself with a knowledge of production, the methods of machining, the costs of manufacturing and the possible commercial limits of machine operations. He should take a course of training in the factory or, through a close cooperation with the production manager, gain step by step a general knowledge of production requirements and of the instincts of commercialism.

The production manager also can learn from the engineer facts that will make him a more valuable factor in his organization. We all have heard the remark made by the production department to the effect that, if the engineering department had used an ounce of horse sense and got down to brass tacks on such-and-such a design, profits would have been made instead of having a receiver at the helm. In all of this, there may be some justification. Many business failures have been defined by A. R. Erskine, president of the Studebaker Corporation of America, as follows: "First, the lack of simplicity and directness in methods; second, to rely too much on theory and guesswork and too little on facts."

"Cooperation" an Essential Slogan

Business policies or decisions based on pure opinions lead to definite trouble that a direct diagnosis and cooperation on the part of departmental heads would avoid. No business can be run unless all departments have the essential facts in hand. To conceive, to design and to invent are simple enough for those who are technically trained but, if we are not in accord with the fundamentals of production, simple and direct in our administrating and manufacturing, we have failed as progressive engineers of the times. We should exercise caution and cooperation before taking action on engineering designs, and our nearest co-partner should be the production manager.

The time to gain his confidence and to benefit by the cooperation and ability of the production manager is not after a design has been completed and put into production but at the time of its inception. Let him criticize, tear apart and make suggestions. With the proper spirit of cooperation that is essential in any well-regulated organization, we will be surprised to find that brains are brains whether in the head of an engineer or that of a production manager. The combination of these brains, working in close cooperation in an organization, will result in the saving of millions of dollars to industry that are now wasted. Every design, every detail and every drawing that is not the subject of study and discussion with the production manager adds its small or large quota to the already large waste of materials, effort and money.

When we stop to consider what the saving of a pound of material on a volume-production basis means, we can grasp the necessity for this close cooperation between production and engineering. The slight changing of a pattern, the adding of a boss, the changed location of a drilled hole, all simple operations and appearing to be of little consequence, may constitute the difference between a profit or a loss on a product. The expense of tooling, of manufacturing and of assembling are determined by the design. Therefore it is to the advantage of a manufacturer to study carefully his engineering and production departments and see that they live in the same house. The engineer, as the creator, should have the confidence and the enthusiasm of the organization. He should supply all departments with energizing power; he should be the dynamo. The production manager, as the producer, should be the brakes and the conservatism of an organization. He should be the motor and not the dynamo.

Last fall the Society held a Production Meeting at Detroit, the first of its kind held by the Society and probably the best attended professional meeting of the Society. The second Production Meeting will be held at Cleveland this month, and this meeting is drawn to your attention to show that the question of production on a quality basis is having the attention of engineers. Efforts are being made to draw the production and the engineering departments into the close harmony and cooperation that is desirable if we are to serve the public honestly and to the best of our ability.

Future Demands

I have endeavored to point out briefly the responsibilities of the engineer and of the production manager and to show that the two positions should constitute a close working, harmonious unit. That this condition does not exist, in many cases, is demonstrated by the changing requirements of the engineer. He is being urged to take up the problems of manufacturing and production.

In many manufacturing organizations the production engineer is becoming a reality. He is the technically trained man who steps into manufacturing with a double-barreled knowledge at his command, that of engineering and manufacturing. In some cases the production engineer is an intermediate between the manufacturing and the engineering departments, the man within the organization who makes final decisions on engineering, manufacturing and quality. Where we find the production engineer in this position, we will find an organization slightly out of tune. Cooperation probably exists, but the question of the volume of production is uppermost and probably is demanded of the production manager; this results in a lowering in the quality of the product in the anxiety to meet the production schedules. The production engineer in this organization fulfills a most valuable service. He is familiar with the engineering requirements of the product; he sits in judgment on the quality of workmanship and manufacturing that is to be acceptable as the finished product; he is engineer, in-

spector and manufacturer combined in one. But he is essential only in the larger volume-production organizations where the duties of the engineer and the production manager are taken up entirely with problems of engineering and manufacturing.

The engineer, the production manager and the production engineer are each an essential to industry. They are dependent upon each other to a greater extent than are the heads of other departments. Therefore it is the duty of each of these other heads of industrial departments, oftentimes, to smother his pride, to give and take, to work hand in hand and to cooperate so that industry can progress in economy, quality and service to all the peoples of the earth.

TESTING OF MATERIALS AND ITS EFFECT ON ENGINEERING

(*Concluded from p.* 300)

WHITNEY SLAGHT:—Would it be possible to use the same methods for testing aluminum alloys as for alloys of steel?

PROFESSOR UPTON:—So far as fatigue testing is concerned, we expect to work them in the same way.

MR. SLAGHT:—Then the properties of aluminum alloys do not require a different method of testing than those of steel?

PROFESSOR UPTON:—If they are intended to withstand the same stresses as steel, they should be tested in the same way.

W. R. GORDON:—What does Professor Upton consider the proper uses of aluminum bronze? What are its possibilities as a bearing-metal?

PROFESSOR UPTON:—Do you mean a 10-per cent aluminum bronze? This will have a tensile-strength of about 100,000 lb. per sq. in.

MR. GORDON:—Yes. Would you consider it a proper metal for worm-gears?

PROFESSOR UPTON:—I would not like to say. I know that it is being tried out. The only sure way of finding out is by testing. No abrasion machine produces results that we do not question; so, the only way is to test the material in service.

MR. MILLER:—Do you prefer the Rockwell to the Brinell test for hardness?

PROFESSOR UPTON:—The Rockwell test is the same in scheme as the Brinell, and differs only in details that rather favor the Rockwell. The Rockwell has a diamond point instead of a steel ball and makes a small impression of about the same size as does the scleroscope hammer. Sometimes the Brinell ball deforms before the material that is being tested does. The Brinell principle is correct and has been worked out with a diamond ball. We cannot afford diamond balls, but we can afford small diamond points. The Rockwell machine does a thing that the Brinell does not do. It puts on a load of 10 lb., then a 100-lb. load, and then goes back to the initial 10-lb. load. The gage is read with the initial load and is checked when it comes back the second time. You get definite contact with a piece. The shape of the conical impression is always the same, regardless of its size. You do not get this with the Brinell machine.

MR. MILLER:—We find the Brinell test very satisfactory for steels of moderate hardness, but it is not so good for very hard steels.

PROFESSOR UPTON:—With the Rockwell instrument, you can go up through all the harder steels. This is partly due to the shape of the diamond, which is conical; whereas, in the Brinell test, a steel ball is used. It is also partly due to the initial load of 10 lb.

Hydraulic Four-Wheel Brakes for Automotive Vehicles

By Malcolm Loughead[1]

Detroit Section Paper *Illustrated with* Drawings

FOUR-WHEEL brakes actuated by hydraulic pressure were described and discussed in a paper presented by Mr. Loughead before the Detroit Section in May, 1923, and the paper, with additions, was to have been presented at the Semi-Annual Meeting at Spring Lake, N. J., in June, 1923. Since Mr. Loughead was unable to attend the latter meeting and present his revised paper, the only discussion by members was that offered at Detroit, and it follows the revised paper printed herewith.

ABSTRACT

FOLLOWING a discussion of the braking ability of rear-wheel brakes only, front-wheel brakes only and brakes on all four wheels, the author answers citations in disfavor of four-wheel brakes and compares external with internal brakes in favor of the external type.

The hydraulic four-wheel brake system is illustrated and described in detail, and a statement is made of four-wheel-brake design requirements. These are: Reliability, equalization, stopping ability, control, simple adjustments, minimized service requirements and provision against wear. Each requirement is then sub-divided and given separate consideration, claims being made in regard to how well the hydraulic system meets each requirement, with reasons therefor. A summary of the main advantages of the hydraulic compared with the mechanical type of four-wheel brake is made in conclusion.

Whether automotive vehicles should or should not be equipped with brakes on the front as well as on the rear wheels is a subject of vital importance to the entire industry. The number of deaths caused by automotive vehicles in the United States alone is appalling, especially when we acknowledge that a very high percentage of these accidents could be eliminated if all automotive vehicles were provided with adequate and properly designed braking facilities. Most of our tests of four-wheel brakes have been made at a car speed of 30 m.p.h. but, at this speed, only a small percentage of present-day automobiles equipped with conventional two-wheel brakes can be stopped within a distance of 130 ft., even when these brakes are adjusted and lubricated properly.

Disadvantageous Features

Two points commonly cited against the usage of four-wheel brakes are the possibilities of locking the front wheels and of rear-end collisions caused by a too sudden stoppage of the car, with the failure of the car behind to stop likewise. If front-wheel brakes are not designed properly, they may lock the front wheels and cause a serious accident due, among other possibilities, to radical interference with the steering ability. My experience is that it is advisable to make the front and the rear brakes of the same design and size; otherwise, if the rear brakes are larger than those in front, there will be a sacrifice of stopping ability under dry-pavement conditions. If the front brakes are larger or more efficient than those at the rear, they will lock the front wheels, especially when the coefficient of friction of the pavement is low. No accidents due to the locking of front wheels of cars equipped with our four-wheel brakes have been reported.

Danger from rear-end collisions lies more in theory than in actuality. They have occurred in a few instances, but the resultant damage has been slight. Damage to lamps and radiators is more expensive than that done to rear tires and fenders, but the car that follows another usually runs 15 to 30 ft. behind and has that much additional space within which to stop.

External versus Internal Brakes

With regard to the expansion of brake-drums on account of heat generated by friction, the external has a decided advantage over the internal type of brake. When the internal type is used on a long grade, when several successive and severe applications of it are made and especially when the foot-pedal adjustment is very close to the toe-board, the expansion due to heat may reach an amount such that the car is virtually without brakes although, with the same adjustment, the brakes may function perfectly when cold. The opposite condition obtains with the external type; the hotter the brake-drums become, the greater the available pedal-travel will be, a point often overlooked in emergency-brake design and one that can cause a serious accident if a car having hot brake-drums is left standing on a hill.

Oil leakage from the rear axle and over-greasing of front wheels commonly cause internal brake-shoes or bands to become saturated with oil and grease so that the car is practically without brakes, but this is much less likely to occur with the external type of brake. Distortion as well as expansion must be provided against for brakes of the internal-shoe type, and it is very difficult to attain as high a ratio of pedal movement to brake-shoe movement for a given pedal pressure as can be realized with the external type.

To design a brake-drum that will not distort is largely a matter of weight and cost. My experience is that the cost of a drum built rigidly enough to withstand the pressure of internal shoes without becoming distorted will be triple the cost of a pressed drum suitable for external brakes.

The Hydraulic System

Pressure in the hydraulic four-wheel-brake system is built-up in a cylinder mounted on the transmission case near the fulcrum of the brake-pedal. Fig. 1 shows the layout, including the master cylinder, the piping and the reservoir tank. A piston in this cylinder is connected directly to the lower end of the brake-pedal forging, as shown in Fig. 2, and is forced into the cylinder

[1] Chief Engineer, Hydraulic Brake Co., Detroit.

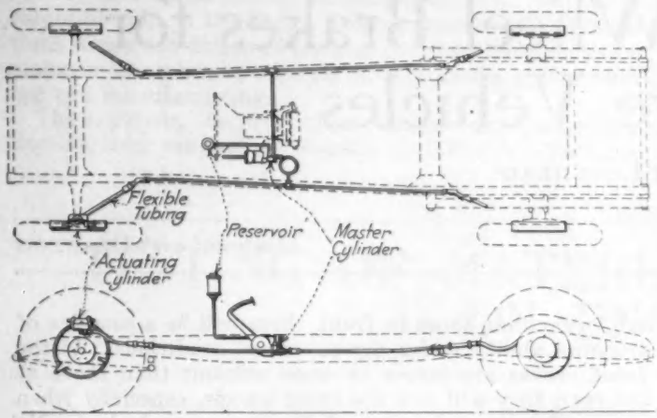

FIG. 1—LAYOUT OF THE LOCKHEAD HYDRAULIC FOUR-WHEEL BRAKING SYSTEM SHOWING THE FLEXIBLE TUBING THROUGH WHICH OIL IS FED TO THE ACTUATING CYLINDERS

by the depression of the brake-pedal itself. The ratio of the movement of the pedal to that of the piston is about 4 to 1. Four copper pressure lines or tubes lead from the master cylinder to points on the chassis frame adjacent to each of the four wheels, as indicated in Fig. 1.

Mounted rigidly on the dust shield or anchor bracket at each one of the four wheels is a brake-band-actuating cylinder that is connected to the pressure line on the chassis frame by a suitable length of seven-ply rubber hose capable of resisting a pressure of 2000 to 2500 lb. per sq. in. before bursting. A section of the hose assembly is shown in the lower portion of Fig. 3. The hose has a close-wound coil-spring inserted while the hose is under a pressure of 1200 to 1400 lb. per sq. in. There is practically no expansion loss up to a pressure of 1000 lb. per sq. in., this being a very important factor with hydraulic brakes because two movements, those of steering and of spring action, must be provided for by using flexible connections. An expansion loss in the hydraulic system causes a lessening of the amount of pedal travel; to compensate for this loss it is necessary to reduce the amount of pedal leverage and this would require a greater pedal pressure.

The copper tubing used for the pressure line on the chassis is supported by clips inside the channel-iron of the frame and terminates opposite each wheel in a standard union nut or coupling. We have never had a failure of brake action due to copper tubing when it was used as a non-flexible member. In some of our first installations we used a coil of copper tubing at each of the brake-bands to take-up the movement, but leaks developed due to the flexing of the tube. The present tubing is not allowed to flex, and I attribute the absence of tubing trouble also to the support of the tubing by the union nut instead of by the flared end of the tube. I have had three failures of the gasoline pipe-line, and each showed a spiral crack at the base of the flare of the copper tubing.

BRAKE-ACTUATING CYLINDERS

Each of the individual brake-actuating cylinders on the four brake-supports contains a pair of opposed pistons, one at either end of the cylinder. The brake-actuating-cylinder assembly is shown in Fig. 3 also. These pistons act against two levers that are, in turn, connected to the brake-band ends, the brake-bands being of the external contracting type. Liquid is admitted to these brake-actuating cylinders between their opposed pistons through an opening at the cylinder centers. When the brake pedal is depressed, the opposed pistons in each cylinder are forced apart by the hydraulic pressure set-up in the system by the master cylinder; one end of each of the levers in each brake-actuating cylinder bears on the head of one of its opposed pistons and the other end exerts a pull on the brake-band end, drawing it tangent to the drum. Smooth brake action is assured by drawing the brake-band ends in a line tangent to the brake-drum and not allowing the ends to "snub." Since the opposed pistons are forced apart by hydraulic pressure, it follows that the pressure on both pistons in all four brake-actuating cylinders must be equal. The brake-actuating-cylinder group-assembly is shown in detail in Fig. 3.

LIQUID COMPOSITION

A mixture of alcohol and glycerine, 40 per cent alcohol and 60 per cent glycerine, constitutes the liquid used in the hydraulic braking system. Any leakage of liquid from the system is compensated for from a small reservoir mounted on the dash that replenishes the master cylinder automatically. When the master-cylinder piston is in the "off" position, it uncovers several small port-holes, each about 0.02 in. in diameter, that communicate directly with the reservoir. This allows a free flow of liquid from the tank to the master cylinder and replenishes any loss of liquid that may occur during the application of the brakes. If there is any loss of liquid, it is very slight; in fact, we have run cars for several months with a closed line and have lost practically none of the fluid. Rawhide is used to pack the pistons and prevent any leakage of the fluid from the hydraulic cylinders.

FOUR-WHEEL-BRAKE DESIGN

In the design of a thoroughly satisfactory four-wheel braking-system, the following six factors are of prime importance;

(1) Reliability
(2) Equalization
(3) Stopping ability
(4) Control
(5) Simple adjustments and minimized service requirements
(6) Wear and replacement of parts

For reliability, three conditions are essential; (*a*) a high factor of safety throughout the system, (*b*) that all movable parts function as designed and that the least

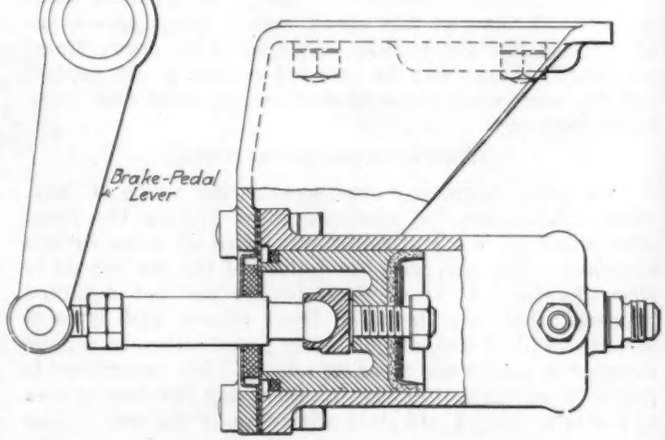

FIG. 2—MASTER CYLINDER ASSEMBLY

possible number of orifices for their lubrication be included and (*c*) that satisfactory performance be attained during all kinds of weather conditions.

A high factor of safety is provided in the flexible hose, the weakest part of the system. Its bursting pressure is 2000 to 2500 lb. per sq. in. and the maximum pressure to which it is subjected in an emergency stop is 300 lb. per sq. in. The pedal clevis-pin is the only movable part that requires lubrication; it is well protected under the floorboards and is accessible. The alcohol content of the alcohol-glycerine fluid can be increased for low-temperature conditions; hence, the system is independent of severe low temperatures.

Equalization

Proper equalization of four-wheel brakes demands (*a*) that the application of the brakes should have no effect on steering ability; (*b*) that it should prevent skidding; (*c*) that it should prevent locking of the wheels, especially the front wheels, and (*d*) that it should not reduce the braking effectiveness under varying road conditions.

Steering ability is not affected in the hydraulic system because the pressure of the liquid is the same in all parts of the system and perfect equalization obtains up to the brake-band ends. Variations in the coefficient of friction of the brake-linings, such as may occur when one front brake becomes wet, tend to make that brake more effective than another. By locating the center-line of the knuckle-pin so that its projection is ½ in., or less, from the center-line of the tire on the ground, the rise in the coefficient of friction of the saturated brake is not noticeable and no effect is transmitted to the steering-wheel even at high car speeds.

As a means of preventing skidding, the four-wheel brake is efficient only to the degree of equalization attained. Positive equalization is secured up to the brake-band ends in the hydraulic system. This claim of positive equalization has been questioned by engineers who say that any great variation in the coefficient of friction of the brake-lining would offset the admitted equalization up to the brake-band ends. For the most part, such variations in the coefficient of friction result from heat generated during a long-continued brake-application and would affect each brake proportionately. Therefore, the change in the coefficient of friction cannot be said to affect equalization at each of the wheels but, rather, would result in requiring a greater pedal effort to accomplish a given result. Assuming that a change in the coefficient of friction is not distributed equally to the four wheels, it is none the less desirable and necessary to maintain equalization in the operating mechanism, unless one argues that two wrongs make a right.

Some means to release the brakes somewhat, at the time the wheels stop rotating, is necessary to prevent locking of the front wheels without reducing braking effectiveness under varying road conditions. Some experiments have been made by others, and we have followed European attempts to accomplish this release of the brakes, one being to have each wheel equipped with one or more weights, the centrifugal force of which tends to release the brakes as the car speed decreases. But this is not commercially practicable and we have found no other way of incorporating the principle with a successful brake.

Stopping Ability

Some engineers have advocated a longer rather than the quickest possible stop and have contended that abrupt stops will result in frequent rear-end collisions and much property damage, but a questionnaire answered by 1000 users of the hydraulic four-wheel brake reports very little damage of this character and proves it to be a matter of education. A car driver appreciates the security afforded by this type of brake but will not subject himself and his passengers unnecessarily to the inconvenience resulting from a very sudden stop.

TABLE 1—TESTS OF STOPPING ABILITY OF HYDRAULIC FOUR-WHEEL BRAKES

Model 57, Cadillac Phaeton Car—Speed, 30 m.p.h.		
Total Weight, with Driver, lb.		4,400
Weight on Front Wheels, lb.		2,000
Weight on Rear Wheels, lb.		2,400
Tires	Make	Mason Cord
	Size, in.	35 x 5
	Pressure, lb. per sq. in.	65
Road Surface		Dry Concrete

Stopping Distances; Average of Five Stops.		Ft.
Brakes Applied on	Rear Wheels Only	89.2
	Front Wheels Only	67.6
	Front and Rear Wheels	36.8

Tests of the stopping ability of the hydraulic four-wheel braking-system were made recently, the results being stated in Table 1. A pistol was mounted rigidly on the chassis of the car; its trigger was connected to the brake pedal and set to fire a bullet into the road surface when the pedal had moved 1 in. After the car had been brought to a stop, the distance from the muzzle of the pistol on the car to the imprint of the bullet in the road surface was measured with a steel tape. The stopping distances stated in Table 1 are an average of five stops in each case, and all stops were made from a car speed of 30 m.p.h.

With a car fully loaded, our tests show that it is necessary to provide 1 sq. in. of brake lining for each 12.5 lb. of car weight. If the ratio of the car weight to the area of the brake-lining is increased to more than 16 lb., the surface of the lining may burn and cause a decrease in the coefficient of friction.

The ratio of 12.5 lb. of car weight to 1 sq. in. of brake-lining conforms fairly well with standard practice. Considering a car that carries seven passengers, weighs 5360 lb. and has 3200 lb. of this weight on its rear wheels, we find the average area of the brake-lining to be 225 sq. in., approximately; this gives 14.5 lb. of car weight

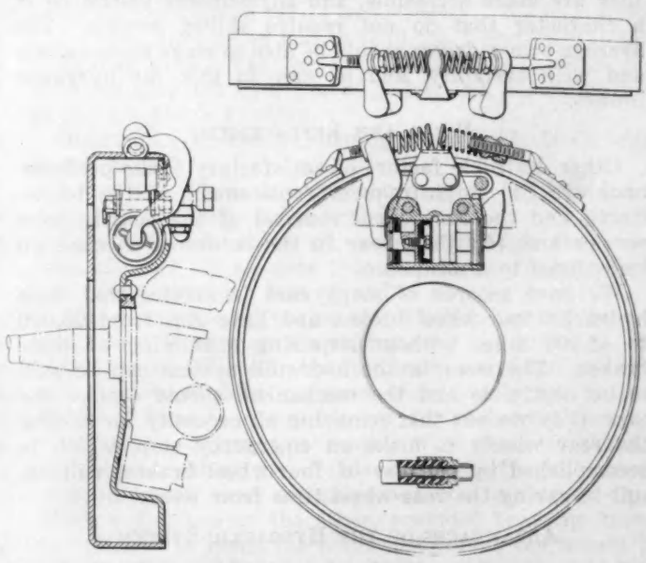

FIG. 3—WHEEL CYLINDER GROUP ASSEMBLY WITH A SECTION OF THE HOSE END SHOWN IN THE LOWER PORTION

on the rear wheels for 1 sq. in. of brake-lining. Owing to a shift of the center of gravity of the car when brakes are applied, this car weight per square inch of brake-lining area will be somewhat greater on the front wheels when the brakes apply to all four wheels or to the front wheels only.

Control

Adequate brake control requires (*a*) light pedal-pressures, (*b*) positive release, (*c*) a minimum length of pedal stroke and (*d*) a constant pedal-pressure for any given stop.

Because of the low friction loss in the hydraulic means of brake application, the high friction loss of the mechanical hook-up is avoided and it is therefore possible to use only a moderate pedal-pressure without employing any servo-mechanism. A positive brake release is accomplished because there are no mechanical joints or bearings to freeze or seize. The brake members are returned positively to their normal position after the release of pressure by an adequate return spring. A minimum length of pedal stroke is permitted because there are no built-up losses of motion due to a multiplicity of mechanical joints. Each inch of pedal stroke is productive of work at the brake-band ends. The pedal pressure remains constant for any given stop, except as it may be affected by variations in the uniformity of the brake-lining and this, in our experience, is not a large factor, because of the effectiveness of the hydraulic actuating means.

Simplicity of Adjustment

Desirable adjustment features that minimize service requirements are (*a*) simplicity, (*b*) infrequency and (*c*) accessibility.

No adjustment for brake equalization is required by the hydraulic system; the inherent equalization in the actuating means is complete throughout so long as the friction members are functioning. Adjustments for wear will be infrequent if a liberal area of brake-lining is provided by the design, say 1 sq. in. of lining for each 12.5 lb. of car weight. Cars equipped with hydraulic brakes have been driven 8000 miles without adjustment.

Regardless of the simplicity of adjustment that is provided, many users will neglect adjustments unless they are made accessible, and adjustments should be of a character that do not require skilled service. The average owner-driver should be able to make them quickly and with certainty, and he can do this for hydraulic brakes.

Wear and Replacement

Other desirable factors of satisfactory brake performance are (*a*) an infrequent replacement of the brake-lining and the consequent removal of the vehicle from service and (*b*) that wear in the actuating mechanism be reduced to a minimum.

We have records of many cars in service that have hydraulic four-wheel brakes and have run from 25,000 to 48,000 miles without requiring a relining of their brakes. The wear in the hydraulic system can be said to be negligible and the mechanism should outlive the car. It is obvious that removing all necessity for sliding the rear wheels to make an emergency stop, which is accomplished by the use of four-wheel brakes, will result in saving the rear-wheel tires from wear.

Advantages of the Hydraulic System

Inherent equalization is secured in the hydraulic system of braking. Exactly the same pressure is applied to each of the four brake-bands by utilizing a column of liquid under pressure against pistons acting in cylinders adjacent to the brake-band ends, and equalization is thus maintained regardless of brake-band adjustment.

Since there are fewer mechanical joints, the friction is lowered considerably. This advantage becomes pronounced after several months of service when, due to a lack of proper lubrication, brakes of the mechanical type show wear in some joints and seizing or binding in others.

The flexible connection between the brakes and the chassis prevents trouble due to the improperly related movement of a mechanical brake-linkage.

Brake adjustment is not affected by a change of loading in the car. The movement of the car body in relation to the axles when traveling over rough roads does not react against the brake pedal. This reaction does exist in some forms of mechanical brake and is particularly noticeable when coasting down a rough grade because the brakes then have a tendency to lock and release alternately, due to the movement of the car springs. The operation of the hydraulic system is very smooth on the roughest of roads.

Pedal-pressure requirements are moderate in the hydraulic system and, with a pedal stroke of 5 to 6 in. and a moderate pressure, the necessity of utilizing any servo-mechanism is eliminated by taking advantage of the low friction loss, absence of lost motion and external-band brakes anchored to give approximately 200 deg. of brake-lining in contact with the brake-drum on the wrapping end and in the forward direction of rotation.

THE DISCUSSION

QUESTION:—What liquid is used?

ANSWER:—We use an alcohol-glycerine solution, approximately 60 per cent glycerine and 40 per cent alcohol.

QUESTION:—How does the cost of hydraulic brakes compare with that of mechanical four-wheel brakes?

ANSWER:—That depends somewhat on the design of the mechanical brakes. To date we have seen no mechanical brakes that are equal in cost to hydraulic. We can build the toggle-joint system as used on some foreign cars for a little less.

QUESTION:—What about the design of front axles to which four-wheel brakes are applied, as regards the effect on the steering?

ANSWER:—Placing the intersecting lines of the knuckle ½ in. or less from the center-line of the tire does not affect the steering much. We have a car that is running now in which the line is ⅜ in. from the center of the tire; the car when traveling at a speed of from 65 to 100 m.p.h. could be stopped with one wheel completely shut off, practically without moving the steering-wheel.

QUESTION:—In case the operating fluid congeals or freezes in cold weather, what means of operating are provided?

ANSWER:—In extremely cold weather we run the alcohol content up the same as we should with radiators.

QUESTION:—When city ordinances require two sets of brakes, are independent mechanical means of operation provided?

ANSWER:—In that case we recommend an emergency brake, if for no other reason than locking the car when it is standing.

QUESTION:—Is there no emergency brake otherwise?

ANSWER:—None, except either the drive-shaft brake or the internal brake used in conjunction with the four outside brakes, which are operated by the hydraulic system.

QUESTION:—Are the rear brakes designed to take effect slightly ahead of the front brakes, on account of the greater weight at the rear end of the car?

ANSWER:—No; because even though there is more weight on the rear axle, in the ratio of 60 to 40 per cent, as, of course, we all know is the case, in stopping with four-wheel brakes the shifting of the center of gravity of the car makes the braking of the front wheels more effective than that of the rear wheels.

As an illustration of the shifting of the center of gravity, braking the front wheels of a car traveling at 30 m.p.h. will stop the car in 67 ft. while braking the rear wheels will stop it in 89 ft., which means a decided shifting of the weight; the rear wheels tend to slide before the front wheels even though more weight is at the rear.

QUESTION:—What would happen if a connection or a pipe should leak or break?

ANSWER:—The same thing that would happen if the threaded end of a brake-rod should break, a contingency that is not uncommon. Our system of piping has shown no breakage, and I should say that the percentage of failures would be no higher than that of the threaded ends of brake-rods. You would have to use a brake in either case, whether mechanical or hydraulic.

QUESTION:—Would there be an indicator on the reservoir to indicate that a leak had occurred in the system?

ANSWER:—No; if there were a leak in the system sufficient to cause a failure, the pedal would not remain in position but would drop to the floor-board when it was applied.

QUESTION:—Are the torsional strains on the axles greater or less when 32 x 7-in. tires are replaced by 35 x 5-in. tires?

ANSWER:—The coefficient of friction is the same whether it be distributed over 10 or over 2 sq. in.

QUESTION:—What effect has front-wheel camber on front-wheel brakes?

ANSWER:—It is taken into account only in designing to make the intersecting line of the knuckle-pin come within ½ in. of the center of the tire.

QUESTION:—Can you give the weight that was added to the Cadillac front axle in the test referred to?

ANSWER:—The axle itself was designed to take the torque load as well as the bending load of the weight of the car. The axle is 12 oz. lighter than the former Cadillac axle; I have no exact figures as to the weight of the drums and the rest of the mechanism. The drums weigh approximately 20 lb. each; the bands and the rest of the mechanism would weigh possibly 15 lb.; between 70 and 100 lb. on a car of that weight.

QUESTION:—Do you find that front axles at present are ample in most cars to withstand the torque of front-axle braking?

ANSWER:—No; they have not a sufficiently high factor of safety. We recommend a better torque-section from the spring pad to the outer wheel or the knuckle.

QUESTION:—Can an internal brake be designed to wrap internally?

ANSWER:—Yes; but not with the same efficiency as an external band. We have shown on tests that when an external band is operated by a straight pull between the gap ends, that is, in a straight line, and an internal brake is used on the same drum and expanded in a straight line, the external brake develops 35 lb. and the internal brake 25 lb. In other words, the efficiency of the two is as 35 to 25.

W. C. KEYS:—Were the bands anchored at the middle in each case?

MALCOLM LOUGHEAD:—They were anchored 180 deg. from the center of the gap.

QUESTION:—How much stronger, approximately, in per cent must the front axle be with front-wheel brakes?

ANSWER:—To give a torsional section. The bending section, of course, is still there and is sufficiently strong.

QUESTION:—Would air in the system have any effect on the action of the brakes?

ANSWER:—It would mean a loss of pedal travel; if there were sufficient air the pedal would go clear to the floor-board without actuating the brakes, so that we take practically all the air out, that is, within a very small fraction of 1 per cent.

QUESTION:—What effect have very cold and very hot weather on operation?

ANSWER:—There is slightly sluggish action with the fluid that we use when the weather gets below zero. This can be counteracted by increasing the alcohol content, which will bring the action back to normal.

QUESTION:—Is tire mileage increased by using four-wheel brakes?

ANSWER:—Yes, considerably. There is practically no necessity for ever sliding the tires with four-wheel brakes; this happens many times a day with two-wheel brakes in traffic, without taking into consideration the conditions in mountainous districts.

QUESTION:—In the ratio you gave of 12½ lb. of car weight to 1 sq. in. of brake surface, is the brake surface the effective area or the total area of the brake-lining?

ANSWER:—The effective area; that is, the area after deducting approximately 4 in. of lining at the anchor-pin point on a 15-in. drum.

QUESTION:—What change, if any, is made in the front springs?

ANSWER:—None. We have never had a spring break. The spring makers have recommended that no change be made when four-wheel brakes are used. The front springs are as strong as the rear springs that take the torque loading in the Hotchkiss drive and the brake loading also.

QUESTION:—Are there two points of adjustment on each wheel?

ANSWER:—There are three on each wheel. One on each end of the band and one at the anchor-pin.

QUESTION:—What is the percentage of loss of liquid?

ANSWER:—That question is very hard to answer because the percentage varies considerably. We have never had a loss, have never had to refill the 2-pint tank in less than 6 months. The car I have mentioned, which has been running for over 20,000 miles, has shown no loss at all in a year's running.

QUESTION:—Is there churning of the oil? If so, what is the effect?

ANSWER:—There is some churning of the liquid, but it has no ill effect on the brakes.

QUESTION:—Is there oil leakage through the coiled wire tubing? If so, does this not rot the rubber casing and cause excessive loss of oil?

ANSWER:—The glycerine and alcohol has a preservative effect, if any, on the rubber.

QUESTION:—What is the minimum weight of car that can be economically equipped with four-wheel brakes? Would a Ford be a suitable car for four-wheel brakes?

ANSWER:—All cars, no matter what their weight may be, whether they are Fords or 6-ton trucks, can be stopped in the same distance; one can be stopped in as short a distance as the other, provided that the torque is sufficient to reach the point just before the wheels begin to slide. All cars should be equipped with four-

wheel brakes. A Ford car that hits a child will do as much damage as a 6-ton truck could do. Neither could be stopped in a shorter distance than the other.

QUESTION:—What is the comparative efficiency of four-wheel hydraulic brakes and the commercial two-wheel type of brake on wet pavements?

ANSWER:—It is very hard to give any figures for efficiency, but four-wheel brakes almost totally eliminate the tendency to skid. When I say tendency to skid I mean the tendency of the car to pivot. The four wheels naturally will slide, but as a rule, they will hold to a straight line.

QUESTION:—Is it true that equalized pressure assures equalized braking effect?

ANSWER:—Any hydraulic system must remain in equalization up to the end of the bands. The only variable to consider is that of the coefficient of friction of the brake-lining itself.

QUESTION:—What is the maximum change in the coefficient of friction in the lining throughout its life?

ANSWER:—That will depend to a great extent on the lining. The best lining we have found, in practical use and on the test stand, has shown a variation in coefficient of friction of approximately 0.20 to about 4.46; and this variation usually takes place during wet and dry conditions of the lining.

QUESTION:—Will the brake-drum cool as well with external as with internal brakes?

ANSWER:—Yes, with a pressed-steel drum. Of course, the ribbed drum, whether cast or forged, will cool more rapidly with internal brakes, but with a pressed-steel drum it is about the same in either case. I might add that the drum radiates through almost double the surface that it does with two-wheel brakes. That question really has reference to four-wheel brakes.

QUESTION:—Water on the surface of the lining usually affects the operation of brakes detrimentally. Does not this show more with external than with internal brakes?

ANSWER:—In our experience, water, mud, or anything of that sort gets into the lining of the external-type brake in less time than it would get into the lining of the internal, but there is one point that offsets this difficulty; it gets out quickly, too. The internal brake will trap grease or oil or mud and takes a long time to work it out in some cases. I should say that these characteristics will about balance.

QUESTION:—What is the difference between the action of four-wheel brakes and that of the conventional type when a front tire goes flat at high speed, or on a turn, especially with the great drop due to the size of balloon tires?

ANSWER:—If a tire goes flat on a car equipped with four-wheel brakes you can stop in a much shorter distance than you could stop with two-wheel brakes. As a flat tire has a decided tendency toward pulling the car over into the ditch, and as you can stop in a shorter distance, I should say four-wheel brakes were safer.

QUESTION:—How is leakage prevented with hydraulic pistons?

ANSWER:—The cup packing that we are using now is of rawhide construction. The rawhide is the only form of hide-leather cup that will really hold the liquid without leaking. When I say "leaking" I mean that no seepage gets past it. In the process of tanning the rawhide the fat cells are not broken down, but the rawhide is cured with them, so that it is not porous. Usually, when leather is tanned all the fat cells are broken, the gelatine is taken out and the leather is impregnated with wax, which can only be put into the leather at a temperature of less than 135 deg. fahr. Temperatures higher than this would injure the leather. In cold weather the ordinary leather cup gives good service, but in summer the heat forces the plugs of wax out of the leather, causing a gradual seepage at all times. We have absolutely no trouble when we use all rawhide packing.

We are making a search now and expect in a very short while to have a cup packing manufactured from fabric, rubber composition and the like, which would assure a uniform grade and a larger production than will any leather we might obtain. This packing has better possibilities from a production standpoint, though our present packing is giving us no trouble.

QUESTION:—What about the car behind that has rear-wheel brakes only?

ANSWER:—I think that is the business of the driver.

QUESTION:—Some builders of foreign cars are advocating that front-wheel brakes should be operated slightly in advance of rear-wheel brakes. What is the reason for that?

ANSWER:—Our reason for not doing it is that the shifting of the center of gravity always throws more weight on the front wheels, so that the rear wheels have a tendency to slide before the front ones, though the efficiency of the front wheels has not been increased. Many foreign cars with toggle operation of the shoe make this more necessary than does a balanced system of the hydraulic type.

QUESTION:—What minimum clearance do you allow between the lining and the drum?

ANSWER:—A minimum of 0.025 in. This will allow for expansion of the drum up to 650 deg. fahr.

QUESTION:—Does the brake-lining wear better with four-wheel brakes, or worse?

ANSWER:—Very much better. I might cite an instance of a Model-66 Pierce Arrow that weighs approximately 7000 lb. and carries 18 passengers, over the Ridge Road in California, which has a 7000-ft. rise. The rear brakes were relined after 48,000 miles, the front brakes after 32,000 miles of service. This shows the difference in efficiency between the front and the rear brakes on account of the shifting of the center of gravity.

QUESTION:—Is this particular hydraulic brake the type used by the Duesenberg car that won the Grand Prix?

ANSWER:—In principle, yes; but it is not the same in detail.

QUESTION:—What would the braking action be on the front wheels while turning a corner, as compared with straight running?

ANSWER:—If the brakes are properly equalized there will be no difference.

QUESTION:—How is the pressure of the brakes regulated to prevent sliding of the tires?

ANSWER:—That is another condition that varies so greatly with road conditions that it is almost impossible to put on a safety device that will keep the wheels from sliding. If a person puts too much pressure on the pedal it will cause the wheels to slide at times.

QUESTION:—Are the brake adjustments each independent?

ANSWER:—Yes, independent as regards taking up wear, but not as regards taking up equalization. They are inherently equalized at all times.

QUESTION:—How are the adjustments made?

ANSWER:—By two nuts, one on the head of each band extension, and by the conventional anchor-pin adjustment that is used on external brakes now.

QUESTION:—If the front wheels lock, do you not lose steering control?

ANSWER:—Absolutely. If a person puts his foot on the brake-pedal sufficiently hard to lock the wheels, he will lose steering control until he releases it enough to let the wheels revolve again. Cars that we have equipped have run an aggregate distance of 7,000,000 to 10,000,000 miles, with all kinds of average drivers, and have never had an accident on this account. There is only one thing to do, and that is to release the pedal slightly when you feel the wheels sliding.

QUESTION:—Does that mean that the ordinary driver must be specially coached in the use of four-wheel brakes?

ANSWER:—If he gets on a slippery pavement he must always be careful. He should work the brakes intermittently, as he would two-wheel brakes on ice. If he does not work the brake-pedal intermittently he will swerve round; we recommend that the pedal be applied in the same way with four-wheel brakes.

QUESTION:—The public would probably have to be re-educated in the use of four-wheel brakes?

ANSWER:—I should not think it would take much education; in the cars we have put out we have done no more than give warning of that condition and have never had trouble from that cause.

QUESTION:—Do you consider the use of chains on front wheels advisable?

ANSWER:—We have really forgotten what skid-chains are. We never use them any more. I have had one occasion in the last 7 years to use them and that was coming over the Tuscarora Mountains last winter on the ice. I have never used them on front wheels with four-wheel brakes. That includes two trips across the United States, back and forth, under very varied conditions.

QUESTION:—Do you have a reservoir for excess liquid to replace that lost by leakage, and how does it function?

ANSWER:—The tank is mounted slightly above the master cylinder so as to have a gravity feed; it feeds through several port-holes in the master cylinder and replenishes any loss due to the application of the brakes. The loss, however, is very slight.

QUESTION:—Is the band of the brake-lining continuous throughout its length or is a gap left at the rear support?

ANSWER:—We always leave a gap at the rear support, for two reasons: (*a*) it tends to cut out any dirt or foreign matter that may get into the band; (*b*) it is almost a dead point in the lining.

QUESTION:—Would the churning of the liquid cause any chemical change?

ANSWER:—None that we have found to date. I know of no reason for it.

QUESTION:—In making a turn on a wet pavement with the brakes set, is there any tendency to slide off at a tangent?

ANSWER:—Yes, if the wheels are locked. If the brakes are not applied to such an extent as to lock them the steering is normal.

QUESTION:—How often must the packing be replaced?

ANSWER:—We have never had to replace any packing on account of wear.

QUESTION:—Have you tried riveting the lining to the inside of the drum and using a plain steel band of the internal expanding type to obviate the expansion of the drum?

ANSWER:—No, we have not. To get back to the question of wear, on cups, we continue making tests at the factory; during the last 6 months we have had at least 50 cups that have withstood from 300,000 to 700,000 applications of the brake at a higher pressure than would be used in braking. This is equivalent to a life of 7 or 8 years in a car.

QUESTION:—Does the system become inoperative if a pipe line is broken or plugged up?

ANSWER:—Yes.

QUESTION:—Will not the additional margin of safety provided by four-wheel brakes be absorbed in practice by faster driving?

ANSWER:—It certainly should be.

QUESTION:—If the front wheels lock will the car skid, that is, revolve, or will it merely slide ahead in a straight line?

ANSWER:—Unless there is a decided crown or slope to the road, the car will slide in a straight line. There will be a slight side-swerve; if the car is on a crown it will tend to slide sidewise, but not pivot; the car as a whole will tend to go sidewise slightly.

QUESTION:—Is there an advantage in braking front and rear wheels diagonally in two stages?

ANSWER:—To do so would be to acknowledge that the equalization is poor. If the brakes are equalized I think it would be much better to brake all four wheels and not pay special attention to any two. There can be no advantage in braking diagonally unless the equalization is poor.

CLARENCE CARSON:—Mr. Loughead has said that he found the variation in brake-lining coefficient of friction to be from 0.20 to 0.46, I believe. It is rather hard to reconcile that statement with his other statement that four-wheel brakes have uniform equalization. I do not see how, even if he has uniform pressure per square inch on the pistons and the actuating mechanism, he can have uniform retarding of the vehicle on all four wheels, or even on two wheels, if there is that much variation in friction between lining and drum.

MR. LOUGHEAD:—I think that I supplemented the latter statement by saying that we equalized up to the band ends, and that the variable which we could not control was in the brake-lining itself.

MR. CARSON:—It seems to me that, with the greater part of the braking effort on the front wheels of the car and with such a variation in the coefficient of friction, there would be likelihood of bad "slewing" of the car, particularly if the brakes were applied on a curve. I have found that to be so on four-wheel-brake cars that I have tried. Though the cars hold to a straight course when the brakes are applied on a straight road, if the brakes are applied in making a turn, the car is liable to go wrong end first instead of head first.

MR. LOUGHEAD:—That condition does exist to a certain extent, but is within the control of the driver.

MR. CARSON:—It seems to be a problem for the brake-lining manufacturers to produce a lining with a coefficient of friction as uniform as it should have in order to get the results that are sought with this type of brake. At the present time I do not think that any of them can guarantee it.

MR. LOUGHEAD:—As a rule, a decided change in the coefficient of friction is caused by a severe case of heating. That is what I had in mind when I mentioned the extreme drop down Pike's Peak, where the brakes got very hot on account of the 7000-ft. drop in 18 miles. That did not occur with one wheel nor with another; as a rule, it occurs in all four wheels more or less.

MR. CARSON:—The Society has been conducting experimental work during the last 3 years on friction materials, in conjunction with the Bureau of Standards at Washington; and more recently I think there have been several independent investigations. To date I do not

think any brake-lining has been found, regardless of price or make, that will hold to a very close variation of the coefficient. Many of the brake troubles that car builders are criticized for having, are unjust, because from the very nature of the lining and the way it is made, it is almost impossible to produce a uniform coefficient of friction. The problem is to get a satisfactory lining first.

MR. LOUGHEAD:—That is something that is beyond our control in any type of brake, until a better standard of brake-lining is developed. Probably 50 per cent of the poor equalization in mechanical brakes is due to linkage; if that can be overcome, the variables will be reduced from two to one.

MR. CARSON:—I take issue with you a little on that, because these tests have been made after overcoming as far as possible the variation due to brake linkage. They have been made on the drum with the brake-lining applied as directly as possible, and most of them have been on testing apparatus. It has been found that the coefficient of friction of the lining is extremely variable; it will usually start out fairly high on woven types when the lining is new, but the frictional heat developed when the brakes are applied causes the coefficient of friction to fall very quickly. It will recuperate when the drum is allowed to cool. This variation will continue throughout the life of the lining and is caused by different conditions.

Another thing is the possibility of the swelling of the lining. On brakes designed with a very small clearance frictional heat might cause some types of lining to swell and grip. Is it not true that if a front-wheel brake should grip on one side it would be very disastrous?

MR. LOUGHEAD:—No; we have run cars around with one front wheel completely shut off, and there has been no ill effect from it at all. I mean that a three-wheel brake is absolutely practicable from all standpoints in handling the car.

MR. CARSON:—The condition I referred to was of one wheel's suddenly locking.

MR. LOUGHEAD:—Sudden locking is within the control of the driver and depends on the pedal pressure he uses.

MR. CARSON:—The average car-operator will not graduate the first application carefully, but will lock the wheels.

MR. LOUGHEAD:—As I have stated before, I can answer that in only one way; we have 1000 drivers, who have had no trouble whatever in that respect. Of course, 1000 is not a large enough number on which to render final judgment, but we have 1000 drivers who have covered more than 7,000,000 miles with absolutely no trouble from that cause.

MR. CARSON:—We are very much interested on account of its being a frictional material problem, but I think I can see the possibilities of trouble for the brake-lining manufacturers until such time as a lining has been developed that either will be of a sufficient uniformity of coefficient to give the results we are after, or the ratio of braking between the front and rear wheels has been so modified that the "slewing" effect on curves will be overcome, and the public has become so used to four-wheel brakes that the driver behind, in order to save his car, will come to the conclusion that every car ahead of him has four-wheel brakes and will act accordingly. Most of the accidents occur from trying to stop instead of keeping on running. With four-wheel brakes making sudden stops, I can see the possibilities of smashing rear-end collisions, and it will take much slow educational work to offset that condition.

MR. LOUGHEAD:—We expected more of that than has actually occurred. We have had very few accidents. I know of no accident that has resulted in more than a dented fender; when the number of miles that I have stated is taken into consideration, I think the record is remarkable.

MR. CARSON:—A car was brought to me some time last fall, not fitted with hydraulic brakes, however, but equipped with four-wheel brakes, and the statement was made that it could not be made to skid. We succeeded in landing on a man's front lawn rear-end first while steering round a curve; so apparently they do sometimes skid.

MR. LOUGHEAD:—I should say that that was a mechanical-type brake, because it is typical of mechanical four-wheel brakes, but not so in any respect of the hydraulic brake.

MR. CARSON:—Have you had any trouble with the brake-lining's losing its gripping power on account of the lubricant's seeping by the pistons?

MR. LOUGHEAD:—We have no seepage of the lubricant at all; absolutely none.

MR. CARSON:—Have you any cars in operation around Saskatchewan, where the temperature is about 40 deg. below zero fahr.?

MR. LOUGHEAD:—I know of none. The lowest that I know of is approximately 20 below zero fahr. but the fluid we use will not solidify above 60 deg. below zero fahr. if a high enough alcohol-content, which we recommend in that extreme cold weather, is used. We have run cars under test with pure alcohol in the system and have had no trouble on account of it. Of course, it is a little harder to confine and there is more evaporation from the refilling tank under the hood.

MR. CARSON:—Does the high alcohol-content of the fluid affect the rawhide?

MR. LOUGHEAD:—Alcohol has no effect on rawhide at all.

G. L. MCCAIN:—When I wrote my question as to what economical installation could be made on a light car, what I had in mind was that the Ford stops too quickly sometimes. There must be a point at which the light car can be stopped as well as need be with brakes on two wheels. There must be some minimum weight at which it is economical to install four-wheel brakes.

MR. LOUGHEAD:—A car weighing 10 lb. and a truck weighing 6 tons can be stopped in the same distance because the coefficient of friction on the wheels retards them by a given force, so that the weight of a car has nothing at all to do with the distance in which it can be stopped, provided that there is an efficient means of braking the wheels up to the point of sliding; the light car requires it as much as the heavy one.

A. L. CLAYDEN:—Mr. Loughead said he did not know any reason for connecting brakes diagonally except to compensate for poor equalization. That was not the original idea of diagonal braking at all. The original idea of the diagonal connection was this: So long as we had a wheel on each axle rolling free, we would have either end of the car under steering control because it is only when a wheel ceases to roll that steering ability is lost. Consequently if we lock the right front and the left rear wheels, and leave the left front and right rear wheels free to roll, we shall have the locked condition on one wheel on each axle and can still steer the car almost as freely as if no wheels were locked at all.

I was able to experiment several years ago with some of the Argyll cars that were linked up diagonally or linked upon all four wheels. I also experimented a little with a hydraulic system very much like the present sys-

tem but not so well carried out in detail. I also tried four-wheel brakes with diagonal connections and all four together.

The conclusion I reached at that time was that the increased stopping power to be obtained by applying brakes on all four wheels was of much greater practical value than the theoretical advantage of the diagonal connection.

Perrot, the chief engineer of the Argyll Co., is the owner of a number of patents on the mechanical application of brakes. In 1911 or 1912 he was very strongly in favor of the diagonal connection and the Argyll car was built on that principle, with a rather complicated mechanical connection for the brakes.

I had a letter from M. Perrot only recently in which he said he had entirely given up the idea of a diagonal connection and had come round to my original view; that the plus stopping power of all four wheels was worth more than the gain from diagonal arrangement.

L. C. FREEMAN:—Mr. Loughead, I believe, made the statement that the efficiency of his transmission system is higher than that of the usual mechanical linkage. Has he any specific data to support that statement?

MR. LOUGHEAD:—The efficiency of mechanical linkage, properly lubricated, is undoubtedly in the neighborhood of 70 per cent; that is, it transmits approximately 70 per cent of the energy from the foot-pedal to the band ends but is very seldom so high after a car has been on the road from 6 to 8 months. Very few persons lubricate the brake-rod joints until something freezes or sticks and they do not operate; then they begin to drive the pins out, put kerosene on them and so forth. Very few of the cars running today have properly lubricated brakes. They are very inefficient, often not more than 50 per cent of the energy being transmitted from the pedal to the band ends. The efficiency of a fluid column, on the other hand, unless there is seepage, remains constant. The liquid acts as a lubricant on the pistons and the cup packings and is in the line at all times; the lubricant must be there if the brakes are to operate.

MR. FREEMAN:—Constant at approximately what figure? You mentioned 70 per cent for mechanical transmission.

MR. LOUGHEAD:—That is, if well lubricated. I have seen very few transmissions that would average over 60 per cent after they have been in service. An efficient hydraulic system is balanced in all four or five points, whereas with mechanical linkage, by an equalizer bar, each one is added to the other. The efficiency of the hydraulic system is about 82 per cent and remains constant.

MR. CLAYDEN:—The fact has been mentioned that several braking systems developed in England many years ago had been abandoned. Anybody who has had an opportunity of looking at them will have no difficulty in understanding why they have been abandoned.

G. W. HARPER:—Do you find that steering requires more effort at the wheel in making short turns when laterally inclined steering-pivots are used to bring the intersection of the tire and the pivot near together at the ground, than when the steering-pivots are vertical?

MR. LOUGHEAD:—When inclined steering-pivots are used the effort required to turn the steering-wheel is a little greater at very low speeds or when the car is at rest. This is due to the "scrubbing" action of the tire and the slight raising of the car. For several months I have driven a car that has an axle, the stub ends of which are set at an angle of 8 deg. from the vertical and the projected knuckle-pin center-line is $\frac{3}{8}$ in. from the center-line of the tire. This car has never shown any tendency to "shimmy" and the slight increase of steering effort at low speeds is more than offset by the greater ease of steering over rough roads and at high speeds, for most of the road shocks are absorbed by the knuckle-pin and are not transmitted through the steering-knuckle to the steering-wheel.

MR. HARPER:—How do you determine the theoretically correct position of the steering-knuckle tie-rod ball when inclined pivots are used and the consequent length, center to center, of tie-rod balls?

MR. LOUGHEAD:—The only difference between determining the position of the steering-knuckle tie-rod balls with inclined pivots and those of the conventional vertical type is that the pivot center-line is on an angle, and the location of the ball must be taken from a line parallel to this center-line. As the location of the ball is taken from the inclined knuckle-pin center-line, the distance, center to center, between these balls will depend on their vertical location.

MR. HARPER:—If the coefficient of friction between the lining and the brake-drum is subject to such wide variations as from 0.2 to 0.5 under various conditions, what is the advantage of perfect equalization in applying the operating power to the four brakes from the foot-pedal?

MR. LOUGHEAD:—If two wrongs could make a right then we should disregard the equalization of the brake-operating mechanism. As an illustration, take two cars, each of which has the same brake-lining, but in one the operating mechanism is equalized, and the other has very poor equalization. As the braking loss due to the variation in the coefficient of friction is the same in both cases, the difference in stopping ability will be proportionate to the efficiency and the equalization of the operating mechanism. In the design of brakes the variation in the coefficient of friction must be accepted until such time as there shall be a more uniform grade of lining available. When we consider that this extreme change in the coefficient of friction does not occur during more than 1 per cent of the miles traveled, and when it does occur, the change is more or less uniform at each of the wheels, then such change in the coefficient should be considered as a variation in foot-pedal pressure, rather than as a loss of equalization at the wheels.

Trolley Buses and Flexible Vehicles for Street Railway Service

By William P. Kennedy[1]

Metropolitan Section Paper *Illustrated with* Photographs

LIMITATIONS on the space in the September issue of The Journal have again rendered it necessary for a departure to be made from the usual practice of publishing a paper and the discussion following its presentation in the same issue. This paper was published last month and the discussion following its presentation is given below. As is the custom an abstract of the paper precedes the discussion so that those of the members who did not read the paper when it was printed in the September issue of The Journal can gather some knowledge of the subjects covered by reading the abstract if they do not wish to take the time to read the complete text.

ABSTRACT

THE author surveys some of the general conditions prevailing in the street-railway field and the prospective development of a new type of service, in discussing the necessity for closer cooperation between the engineers of the automotive industry and the operating organizations of the railways, the idea being to develop flexible transportation-equipment that will coordinate with the operation of present railway-transportation facilities and to promote the utilization, wherever feasible, of railway power-supply in the employment of flexible bus-type equipment in supplementing and extending railway-organization service.

Changing conditions are outlined, the influences tending toward flexible equipment are stated, and the differences of engineering practice pertaining in the railway and automotive fields are commented upon to show wherein railway and automotive engineers can cooperate.

The trolley bus is considered in some detail as being one solution of the problem, and the Mercedes Stoll, the Filovia and the Max Schiemann systems of operation are described. Present national practices of operation with various types of equipment are mentioned, and a description of dual-power bus-equipment and independent buses having electrical transmissions, with comment thereon, is included. Body and chassis requirements are enumerated, the permanency of public-service transportation and prospective street-railway business is considered, and the requirements of so-called "flexible" equipment are treated at length.

THE DISCUSSION

G. C. Hecker[2]: During the past few years we have heard and read many sweeping statements that the electric-railway transportation-systems of our cities and communities have outlived their usefulness and are about to be junked and replaced by railless vehicles. Many extravagant claims have been made, usually by persons who are either grossly misinformed or who deliberately attempt to distort the facts, that buses can be operated more cheaply than electric-railway cars. Curiously, some of the most distorted statements find their way into the daily press and other publications. As an example, I cite an actual instance.

Recently, there appeared in one of the most conservative and reliable newspapers of New York City a statement purporting to show that the operation of buses is so much cheaper than the operation of electric cars that the latter can scarcely hope to survive much longer. The author presented figures comparing the two types of transportation system that were little short of ridiculous. He estimated the cost of operation of a double-deck gasoline-driven motorbus at 24 cents per mile. As a matter of fact, the Fifth Avenue Coach Co.'s operating cost for the year ended June 30, 1922, was 38.3 cents per bus mile, about 60 per cent greater than the figure used by the author of the article referred to. Incidentally, the author stated that the Fifth Avenue Coach Co. was perhaps the most successful bus system in the Country and he included in his article a statement of its annual earnings, passengers carried and other financial statistics. His comparisons were made without regard to the relative seating capacities or the maximum carrying capacities of the several types of vehicle and, in general, his entire argument was almost devoid of facts or common sense.

As engineers, we are all aware that comparisons made on a bus-mile or car-mile basis must take into consideration the relative carrying capacities of the different types of vehicle. It is sometimes astounding to observe the manner in which so-called experts blandly disregard in their analyses such factors as seating capacity, schedule speeds and other important items that have a vital bearing on operating costs. For the comparison of operating costs of widely dissimilar types and sizes of vehicle, a seat-mile basis is far more accurate than the vehicle-mile. Perhaps the most accurate basis of comparison is the relative cost per passenger carried. This figure, when compared with the fare charged, tells the true story. In this connection, I point out that the cost of operation per passenger carried on the Fifth Avenue Coach Co. system, which operates under extremely favorable conditions, was slightly more than 7 cents for the year ended June 30, 1922, not including fixed charges. Obviously, this company could not exist on a 5-cent fare. Also, the weighted average cost of operation per passenger carried, for a group of 103 city electric-railway companies operating all types of car and under all sorts of operating conditions, was approximately 5¼ cents for the year ended Dec. 31, 1921, not including fixed charges. Similar figures now being compiled for the year 1922 indicate that the cost will be slightly less than this figure.

Bus-Transportation Requirements

The pioneers in the bus-transportation field had very little knowledge of the requirements of the business in which they were engaged. Most of them made little

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effort or else were unable to determine whether or not they were operating at a profit. In many cases the systems of accounting set up did not take into consideration all of the operating expenses, not to mention taxes, interest and customary replacement charges. The result was that many of the lines which showed a book profit actually were being operated at a loss.

Mr. Kennedy has visualized the existing conditions and recognizes the fact that no form of transportation has yet been devised that can supplant the electric railways and the railroads. The railless type of vehicle undoubtedly is destined to play an important part in the future transportation systems of our cities and communities. Its most important function, at present, as pointed out by Mr. Kennedy, is to supplement existing rail-transportation facilities in such a manner that the coordinated service will best meet the needs of the people. It is generally recognized, as Mr. Kennedy points out, that "the existing electric-railway organizations are experienced as no other agency in the utilization and operation of transportation equipment." This view has been widely accepted by state public-utility commissions, many of which have taken the position that the transportation needs of any city can be served best by permitting the established transportation company to maintain a monopoly. In other words, the best interests of a community will be served and transportation will be provided most economically by the elimination of wasteful competition and the coordination of all service under one reliable agency.

In a recent address at Detroit, Alfred Reeves, general manager of the Automobile Chamber of Commerce, said:

> Bus truck-lines have not all been successful generally because of mismanagement or because of routes that furnish insufficient return. All this means that railroad men are the proper ones to take charge of bus and truck operations, making them feeders to the trolleys and steam lines. The automotive industry is anxious to cooperate in every way that will make for each department of transportation finding its proper place, and this means efficiently serving the public

The problem before everyone interested in transportation, therefore, is the coordination of all facilities available. No conflict is being waged between the electric railways and the motor-vehicle industry; instead, the heartiest cooperation exists. Many electric-railway companies are utilizing the motor vehicle today as an adjunct to their rail systems. The trackless vehicle should serve a very useful purpose in the development of territories that heretofore have not been served by any form of transportation, and it should be especially useful in building up new territories until the time is reached when service can be given more cheaply by rail cars. Property owners and residents are assured of a permanent service when the responsibility for that service rests with the electric-railway company, and they will be encouraged to build homes and fully develop the community. It should be borne in mind, however, and the public must be educated to a realization of the fact that if service by buses is more expensive, a higher price will have to be charged for it than is charged on the electric cars.

Preferred Routes and Taxation

A large percentage of the present bus systems are operating over what can be called preferred routes. Naturally, in choosing suitable routes for the inauguration of the bus system, the operators have selected the pastures having the greenest grass. Those who advocate scrapping the railways and substituting buses must not forget that, just as soon as the buses attempt to provide service for an entire community, they will find themselves operating over many lean routes with the result that the net earnings will be seriously affected. This is a condition that practically every electric railway has had to meet. Many routes that are necessary to the transportation needs of a community, if independently operated, will not stand on their own bottoms and must be carried by the profitable lines.

Buses have not been subjected to the same burdens of taxation and regulation as the electric railways have. Most electric railways are required to pay for the paving between the rails and for a certain distance outside of their tracks. They have to pay for watering streets, for snow removal and a percentage of their gross income in state taxes; in many cases, they pay a tax to the community, franchise taxes, license fees and various other charges that put a very heavy load upon them. If the same burdens are placed upon the motor vehicle, it will be found that the railless system cannot successfully compete at the same rate of fare with the electric railway.

Properly Coordinated Service

The possibilities for cooperative work between the automotive engineers and the electric-railway managements in the development of a flexible type of railless vehicle that will meet the rigid service requirements of transportation service have been clearly expressed by Mr. Kennedy, who says: "In the street-railway organization, taken collectively, there exists a greater potential group of consumers than has ever existed in any single direction."

We all must think in terms of transportation and not of certain types of car or bus. We must bend our efforts toward the development of the most efficient vehicles to fit into a properly coordinated transportation service, if we expect to keep abreast of the ever increasing demands made upon us. An interesting point to be noted is that, in spite of the enormous increase in the ownership of private automobiles and the numerous installations of bus systems, the number of passengers handled on the electric railways in 1922 reached what is believed to be the highest level in the history of the industry, approximately 15,000,000,000.

The "store-to-door" freight-delivery business has not yet received any extensive consideration in this Country. It has been developed in a small way by some interurban electric railways through the use of delivery trucks operated to and from terminals. I believe, with Mr. Kennedy, that this field has great possibilities, if the problem is given the careful consideration it deserves.

Design Features

Designers of automotive equipment for passenger-transportation service ought to

(1) Develop designs along such lines as will permit a maximum of standardization
(2) Give adequate consideration in the designs to the inclusion of such safety features or safety devices as have proved practicable
(3) Embody, insofar as it can be economically justified, in the designs such details as will provide for the maximum comfort and convenience of passengers
(4) In the design of cars for city service, give extensive study to entrance and exit design to facilitate the rapid loading and unloading of passengers

(5) In the design of cars to be operated by one man, provide for the satisfactory collection and registration of fares

(6) If practicable in one-man buses, develop apparatus that will enable the operator to call out streets without facing the rear of the car

R. E. FIELDER:—Regarding Mr. Kennedy's statement that the Fifth Avenue Coach Co. used 10 gasoline-electric motorbuses from 1908 to 1914 and would have continued the purchase of this type of equipment were it available and in regular production, these chassis had mounted on them what were then our standard 34-passenger bodies and were later equipped with a De Dion four-cylinder engine of 110-mm. (4.33 in.) bore and 130-mm. (5.12 in.) stroke connected to a generator, this powerplant being mounted on a sub-frame. The power was conveyed through two motors connected by chains to each rear wheel. In first speed, the motors were connected in series; for second speed, in parallel. The acceleration of these buses was sluggish, it being necessary to race the engine to high speed to excite the generator so that enough current could be obtained for the motors to get the bus under way. The gasoline consumption was very much greater than obtained with the standard mechanical-transmission buses and, although the vehicle was similar, except for the electrical transmission, its unladen weight was 664 lb. greater than that of our then standard bus.

Defects in the electrical units were often difficult to locate, even when experienced engineers were called in. Many useful bus hours were consequently lost. After following the history of all the various electric transmissions in this Country and in Europe for the past 15 years, and although substantial improvements have been made recently, mechanical transmissions also have improved and the difficulties formerly experienced with them have practically disappeared. I have no prejudice against either system and believe that the matter resolves itself into a question of which particular system will give the best practical results. In 1916, our experience and records caused us to standardize on buses with the mechanical type of transmission that we are now using. Since we started at this time to build our own equipment, the matter of production was one that was within our own control and not a deciding factor in favor of mechanical-transmission buses. Since 1916, we have tested other modern electric-transmission systems; the result has again justified our decision.

GASOLINE VERSUS ELECTRIC POWER

Mr. Kennedy seems to admit that the electric system, particularly that of the trolley bus, is not all that is needed and that he looks to the gasoline bus to help him out. From my experience, I believe that sooner or later they will discard the electric end of it and take the gasoline bus we are using today. I have visited a number of electric-transportation companies that are using trolley buses in addition to the straight gasoline bus, and have been advised that the latter is by far the more efficient and economical to operate. As feeders, and for filling certain needs, I believe that the former type has its field. However, the trolley car of today also has its field, and it will be a long time before the bus entirely usurps it, if ever. It is significant that five of our customers, who are trolley-car companies, are giving us large repeat orders. At least two of them have found the cost of power for the bus is very much less than the cost of electricity for the trolley car. A company in New York State is finding that the motorbus is more economical to operate than the trolley car.

C. W. KELLOGG[23]:—I emphasize what Mr. Hecker has said about the attitude of the electric-railway companies to the effect that we are trying to take care of the transportation needs of the people who live in the various cities in which we operate and that we are willing to use whatever agencies turn out to be, after proper trial, the best to furnish that transportation.

BUSINESS COMPETITION

Concerning Mr. Kennedy's statement that electric railways have grown up under non-competitive conditions, the natural inference is that, not having had the spur that competition furnishes to any business, they have been inclined to lean back, let events take the natural course and not work very hard. To the outsider that condition may seem to exist, but electric railways have had problems that have served as a spur to great effort. I think they have been just as compelling as the ordinary requirements of competition in other lines of business. For example, the 5-cent fare has necessitated much thought and study on the part of electric-railway management to make both ends meet. Competition in business causes a strenuous effort to improve service and reduce costs, and the competition the electric railways have faced has always been the effort to make a satisfactory financial showing so that, with a fixed rate of fare, they could continue to exist and take care of the rapidly growing and insistent public demands for transportation service.

What has been said about the design of equipment for carrying passengers is most important. In satisfying transportation requirements, the problem that faces automotive engineers more than any other is that of the design of a comfortable, low-center-of-gravity vehicle that the public will find comfort in using.

F. C. HORNER:—I agree with Mr. Kennedy in what he says about the makeshift methods we have used in both chassis and body design of mechanically propelled vehicles for passenger transport. Our automotive engineers have either lost sight of one very important fact or else have purposely evaded it, which is that a motor truck averages between 8000 and 10,000 miles per year, whereas a motorbus averages from 30,000 to 40,000 miles, in some cases reaching 50,000 miles per year, and usually travels at a much higher speed than the truck. Therefore, disregarding the many other differences in operation of the two classes of vehicle, some of which are almost as important as the difference in mileage, we have left much to be desired in meeting the requirements of the passenger-carrying motor-vehicle operator. In this connection, I quote an extract from a paper by H. E. Blain, assistant managing director of the London General Omnibus Co., presented before the Brussels Transport Congress in October, 1922. Referring to the question of motorbus design, he said:

> It cannot be too strongly emphasized that an omnibus chassis is an entirely different chassis from an ordinary truck chassis, requiring special consideration in its springs, its weight and its efficiency, and I was specially struck, during a visit to the United States and Canada in 1921, with the failure there to realize this important point

All of the motorbus men I have talked with abroad referred to this American practice of using an ordinary motor-truck chassis with a few minor modifications as

[23] Engineer, Stone & Webster Co., Boston.

to springs, gear ratio, and the like, with any old makeshift body for passenger-carrying work. However, even in the short time I have been home, I have noticed a very decided change for the better in this matter, and I am confident that we are at last awake to the necessity for a different policy.

The Trolley Bus

Although agreeing with Mr. Kennedy in certain respects on the matter of the trolley bus, I feel justified in calling attention to the fact that the use of the trolley bus in Europe has increased little or none since the installations made at Bradford and at Leeds, England, about 10 years ago. All of the managers of tramway undertakings with whom I discussed the matter felt that the trolley bus was unsuitable for *their* particular city. After expressing this opinion, they usually advised me to go to Bradford and to Leeds, where these vehicles have been in operation for years. In fact, I am told that Mr. Hamilton, the Corporation Tramway's manager at Leeds, was the pioneer in trolley-bus operation in England; he is probably the best informed man on the subject in Europe. Most of these tramway men stated as the principal objection to the trolley bus that it contains many of the disadvantages of the trolley car but, at the same time, lacks most of the advantages of the motorbus. The Bradford and the Leeds trolley buses have had many vicissitudes but, although they are giving satisfactory results in some respects, from the standpoint of economy they leave much to be desired, as is evidenced by the cost figures given in Table 4.

TABLE 4—BRADFORD,[24] ENGLAND, 1921 TROLLEY-BUS AND TRAMCAR COSTS

Items Per Car Mile	Trolley Buses Pence	Trolley Buses Cents	Tramcars Pence	Tramcars Cents
Expense	20.57	42	23.48	47
Traffic Revenue	13.24	27	26.56	54
Profit	....	..	3.08	7
Loss	7.33	15	....	..

[24] In Bradford, 18 trolley buses operate on 9 miles of routes.

In my opinion, the field for the trolley bus in this Country is very limited, for many reasons besides those I have mentioned but which time will not permit discussing now. I should add that a six-wheel trolley-bus is on trial at Bradford, England, and also that the City of Birmingham, England, has recently purchased several trolley buses, the preliminary tests of which I witnessed at Leeds, England, in the summer of 1922.

Dual Power Equipment

One other matter mentioned by Mr. Kennedy is that of using a tractor of the railway company, having dual power equipment, to transport goods between railway terminals and traders' store doors; in other words, to perform a "store-door delivery." This is service by a combination of straight electric-railway lines and trackless-trolley vehicles running through the streets of a city. Is my interpretation of what Mr. Kennedy has in mind correct; if so, will he explain how this would be an improvement over present methods, and also, would it reduce the wasteful traffic congestion in our city streets?

I call attention to the necessity for looking at this whole subject of transportation in the broadest possible way. For, is not the solution of the problem to be found more readily than by simply operating along conventional lines to meet the needs of today? Is it not rather that we should stand off and study the conditions from a distance so as to get the right perspective? An Englishman, who is one of the best informed road-transport men I know, wrote me a short time ago as follows:

> The solution of our traffic problems would seem to lie only in well considered town planning where work places and living places, productive centers and transport facilities will be well coordinated so as to avoid all unnecessary transport, whether of passengers or of goods, and where interdependent places will be knit together by the most direct routes

This, to my mind, sums up the whole problem in an admirable way. I am glad to say that a movement already has been started in New York City to work out the solution along the lines my friend suggests. I refer to the Plan of New York and Its Environs that is being carried on by the Russell Sage Foundation. This will be a thorough and unbiased investigation by experts; consequently, if used intelligently and honestly, the information obtained will, I believe, show us the way to make New York City a desirable place in which to live and carry on our business. At the earliest possible moment, the same remedy should be applied in many other of our cities. I think it is safe to predict that the motorbus will occupy an important place wherever passenger transport is involved.

C. J. McPherson[25]:—From my conversation with a number of representatives, the automotive industry seems to feel that the street-railway industry is not inclined to adopt the automobile as a transportation medium. From my contact with the street-railway industry, that is not true; but the fact remains that, eliminating the three best-known bus installations in the United States, few such installations, if any, are making money where they are operating as corporations. In the Baltimore installation, the cost is roughly 39 cents per bus mile, and the receipts are slightly less than 36 cents per bus mile, showing a dead loss.

A Profit-Earning Vehicle Needed

The street-railway industry must develop or secure a vehicle that can be operated at a profit. The bus apparently cannot be so operated, taking the case I mentioned as typical. If the fare is increased, the total receipts decrease, as has been proved with street cars as well as buses. This has been proved in Boston for the street railways. The trackless trolley has been tried experimentally to see if that is the vehicle. The company I am connected with has installed a large quantity of such equipment in this Country. We hold no definite brief for the trackless trolley, in spite of that. It may be the vehicle and it may not. We are not convinced that it is. The automotive industry has developed a vehicle that is the mechanical marvel of all time, the automobile; but the automobile bus and the trackless trolley apparently are not the vehicles that the street-railway industry needs, and it devolves upon somebody to develop the needed vehicle. It is you automotive engineers who should have the knowledge and the ability to produce that vehicle.

The Storage-Battery Vehicle

Mr. Kennedy mentions the storage-battery too briefly in his paper. Recently, a man very active in the sales end of the motor-truck industry said that the storage-battery vehicle had accomplished so little because of lack

[25] Sales manager of the special works department, J. G. Brill Co., Philadelphia.

of salesmanship, but the gasoline-driven bus has not suffered on that score. If the motor-driven storage-battery truck is practicable, it offers wonderful advantages to the electric-railway industry, which has the power in quantity and at a low price. If that desired vehicle can be produced, it should be operated at a lower maintenance cost than that of the gasoline engine.

Mechanically, the trackless trolley seems to offer wonderful possibilities, but such vehicles as have been produced up to the present are mongrels, using automobile parts that are not particularly well adapted. Automotive engineers should develop parts, primarily axles, that are adapted to that service.

Supplementing a further remark of Mr. Kennedy's, if it is possible to utilize the present power that the street railways have at their disposal by means of the storage battery, it is much to be desired. The trackless trolley, due to its fixed route and its lack of range, has certain political advantages. The fact that the trackless trolley must stay on a fixed route will unquestionably render it easy to secure franchises; it is there and cannot be moved from one locality to another, particularly from one State to another, and it can be taxed and definite responsibility fixed upon it. The information that anyone would need in developing a vehicle for this service is easily obtained. The street railways are ready to give it, either direct from the individual railways or through the American Electric Railway Association. We, as equipment builders to the street-railway industry, stand ready to give to the automotive industry any information we have at our disposal. We are facing a definite engineering problem.

MR. FIELDER:—Our statistics show that the operation cost of the Fifth Avenue motorbuses in New York City is upward of 38 cents per mile. This is due to many reasons, one being excessive taxation and another the fact that we operate two-man buses for the most part. It is a substantiated fact that three of our customers, who operate single-deck buses, are doing so at a cost of from 20 to 24 cents per mile.

PROF. MORTON ARENDT[36]—Mr. Kennedy has emphasized, and correctly so, the need of cooperation between railway and automotive engineers and the consideration of bus transportation and auxiliary bus service by street-railway corporations. Recently, I visited Baltimore and discussed this general situation with the management of the United Railways & Electric Co. This corporation cooperates in the fullest degree. It operates standard trolley cars, buses and trackless trolleys and shows clearly that each vehicle has its sphere. In regard to what New York City has done concerning certain lines of transportation, it must be admitted that emergency demands and therefore temporary routing can only be handled by the gasoline bus, the electric bus or the gasoline-electric bus on account of their extreme flexibility. Each of these self-propelling vehicles has its advocates, but I will not discuss their relative merits now.

Crosstown Service in New York City

Because of the shutting down of certain of the surface transportation lines in New York City and the growing demands for additional crosstown service, Commissioner Grover Whalen of the Department of Plant and Structures of the City of New York was authorized by the Board of Estimate to grant permits for the operation of bus lines by private individuals on a 5-cent-fare basis, so long as the lines conformed to the regulations set up by the Department, and so long as the street-railway companies did not succeed in getting injunctions to stop their operation. The observance of these "City" regulations is made certain by the appointment of "City" bus supervisors. Complaints as regards the returns from these vehicles on a 5-cent-fare basis have not been received by the Commissioner except in a few cases, and these complaints have been satisfied by transferring buses for short periods from paying to non-paying routes and, conversely, to equalize the returns. It can also be emphasized that the so-called non-paying roads are seasonable, being very remunerative in summer. In general, these buses are operated over relatively short hauls; they are well filled and even crowded. When the haul is long, the number of passengers in and out per car mile runs high.

Staten Island Service

I feel that the ordinary gasoline bus is not the solution of transportation in partially undeveloped districts where the law does not prohibit the erection of overhead wires. On Staten Island the City has experimented with the 5-cent-fare emergency-bus and, on account of the hilly country and the relatively long hauls, the operation was abandoned because it did not pay, even before an injunction was issued against the operation of these emergency buses. In casting around for a substitute for bus operation, Commissioner Whalen became interested in the trackless trolley and, in the summer of 1921, decided to construct such a line to cooperate with the municipally operated trolley systems on Staten Island, extending transportation from Meiers Corners to Linoleumville, a distance of about 4½ miles, and from Meiers Corners to Seaview Hospital, a distance of about 1¾ miles. This line was opened to the public on Oct. 8, 1921.

On the Linoleumville line the overhead work consisted of a double-track run; that is, two groups of 00 trolley wires spaced respectively 14 in. between centers and supported by a special insulator manufactured by the General Electric Co. From Meiers Corners a single line, comprising two 00 trolley wires spaced 14 in. between centers, extends to a loop around Seaview Hospital.

After competitive bidding, the single-man-operated trackless-trolley buses furnished for this first line were built under Departmental specifications by the Atlas Truck Corporation of York, Pa. These cars weighed about 12,000 lb. empty, and were designed to have a seating capacity of 32 and a carrying capacity of 50 passengers, although frequently they have carried as many as from 75 to 80 persons. When the cars were first purchased, they were provided with trolley poles of the internally maneuverable type, but it was soon found that such construction was absolutely unnecessary and they were replaced by trolley poles of non-maneuverable type. At first, the motormen found it difficult to maintain the sliding trolley-pole heads in contact with the wires but, after the operators learned the characteristics of the road and the angle of departure from under the wires that the cars could be given, this difficulty was overcome very rapidly.

Construction of the Trackless-Trolley Cars

The first lot of trackless-trolley cars were built on 7-in. channel-iron chassis-frames with 34-in. cushion-wheels and cushion tires. Two 25-hp. motors were suspended in the center line of the chassis frame and were connected to the rear drive by a propeller-shaft that carries a propeller-shaft brake, the drive being of the

[36] Department of electrical engineering, Columbia University, New York City, and Consulting engineer, Department of Plant and Structures, City of New York.

FIG. 21—ONE OF THE FIRST STATEN ISLAND TRACKLESS-TROLLEY CARS ROUNDING A TERMINAL LOOP

worm-and-gear type with a ratio of 8.3 to 1. The motors use from 1.6 to 2.0 kw-hr. of current per car mile with an operating speed of between 17 and 25 m.p.h. Expanding-drum brakes are provided for the rear wheels. The propeller-shaft brake is the emergency brake; the internal expanding brake serves as a foot-brake. The motor controller is of standard series-parallel, KR type, arranged to operate with a foot switch so that the controller cannot be moved unless the foot switch is closed; this latter also serves as a "dead man's" switch. The general practice of trolley-car wiring and illumination is followed in this construction. The tread on the rear wheels to the center line was 56 in. and the height of the floor of the car from the roadbed about 37 in. The City of New York has 22 trackless-trolley buses in operation, 7 of them being the pioneer cars; they were not modified automobiles but were designed from the ground up for trackless-trolley-car purposes.

The specifications that the City issues regarding the turning radius of cars is that the trackless-trolley shall turn within a circle of not more than 15-ft. radius. Cars with a wheelbase of from 140 to 195 in. turn within such a circle very readily after the operator has had some practice.

Fig. 21 shows one of the cars of the first model going around the loop at Meiers Corners. Part of the route of these cars was over a road having very poor surfacing and no roadbed foundations. This was naturally very rapidly cut-up and, as has been expressed by many who have ridden upon it, "it had the appearance of a battle field," as it was full of ruts and holes; in fact, it was almost worthless as a road. A picture of a car operating over this road is shown in Fig. 22.

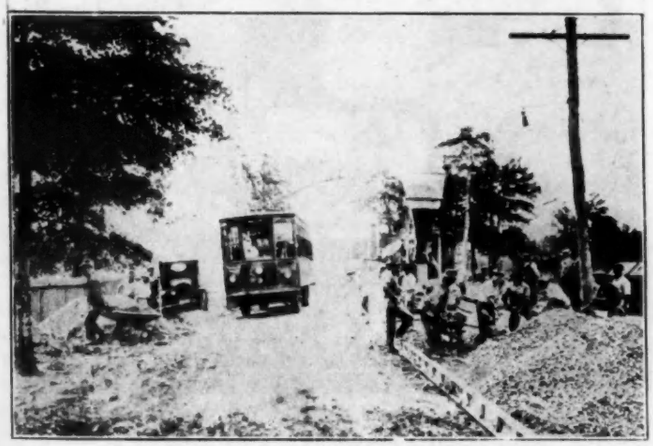


FIG. 23—HOW THE FLEXIBILITY OF THE TRACKLESS-TROLLEY SYSTEM PERMITTED SERVICE TO BE MAINTAINED WHILE THE ROAD SHOWN IN FIG. 22 WAS BEING RECONSTRUCTED

The flexibility of the trackless trolley-bus is shown in Fig. 23 wherein, after the road had become almost impassable for other purposes than trackless-trolley operation, a decision was made to reconstruct it, the trackless trolley cars swinging to one side of the highway so that the road could be rebuilt. This work comprised the entire regrading of this road, the construction of road foundation and concrete surfacing. During this work the trackless-trolley buses operated without difficulty. In the winter of 1921 and 1922 the storms put all transportation facilities on Staten Island except the trackless trolley out of commission several times. Our experience with this pioneer line has certainly demonstrated the unquestionable reliability of this type of transportation over bad roads and in all kinds of weather.

CHANGES IN CAR CONSTRUCTION

Certain inconveniences were observed during the operation of these cars. The bottom of the car was 37 in.

FIG. 22—CHARACTER OF THE ROAD OVER WHICH THE TRACKLESS TROLLEYS OPERATED ON STATEN ISLAND

from the roadway and that required two steps. Second, the tread of the wheels was not wide enough and the cars developed an objectionable side swing. We were instructed to prepare specifications for a new car that would be closer to the ground, eliminating one step, give us a broader wheel tread and bring the springs farther out so that no swaying of the car would occur. These specifications call for a car that has a 195-in. wheelbase and is of the hood type.

Residents of Tottenville, Staten Island, find it necessary from time to time to come to New York City. The distance from Tottenville to St. George is about 19 miles over the highway and probably a little more by railroad, but a round trip costs 80 to 85 cents. The municipally operated trolley-line has a branch that terminates at Richmond, a distance of about 9 miles from Tottenville. In his desire to improve the transportation facilities of Staten Island, Commissioner Whalen, after a careful survey of conditions, decided to install a trackless-trolley line to operate from Richmond, directly at the terminal of the old trolley system to Tottenville. He ordered the installation of this system, so that residents of Tottenville and intermediate places could have transportation for 24 hr. per day at a relatively close headway and low rates to St. George by way of the trackless-trolley extension operated and owned by the City and the old Midland line operated by the City under a contract

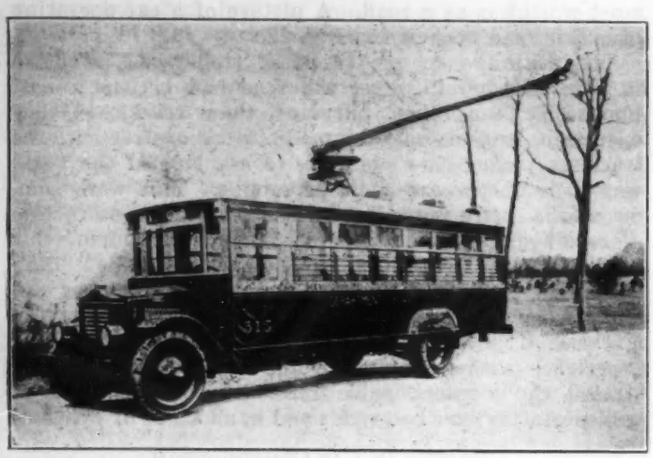


FIG. 24—THE NEWER TYPE OF TRACKLESS-TROLLEY VEHICLE THAT HAS BEEN PLACED IN SERVICE ON STATEN ISLAND

with the Federal Court. Work was commenced in April or May, 1922, and the line was opened to the public Nov. 4, 1924. The same type of overhead construction was employed on this line as on the pioneer Staten Island line, but the type of car was changed, though the same powerplant of 25-hp. motors was retained.

The reasons for changing from the pioneer type of car shown in Figs. 21, 22 and 23, were that (*a*) the distance of the car floor from the roadbed was too great, (*b*) the wheel tread was too narrow, (*c*) the distance between the springs on either side of the car was too little and (*d*) the general appearance of the car was not aesthetic. The specifications were therefore modified to correct these features, and the trackless-trolley bus having the appearance of that shown in Fig. 24 was obtained. The interior of one of these cars, looking forward, is shown in Fig. 25. The accessibility of the motors is indicated by the removal of the trap door. The control resistances, the storage-battery and the emergency lighting-generator are located under the hood as illustrated in Fig. 26.

The car, with its long wheelbase and good springs, rides very well. It is evidently easy to reach the motors or make an examination of the brushes; also, to expose a portion of the propeller-shaft brake so that it can be adjusted from within the car. The KR type of controller is shown on the left at the operator's side, and the cash register is located at the right. These cars are not provided with ordinary rough wooden floors, but with ¼-in. battleship linoleum covering over a tongue and groove floor. We use the emergency lights because at times a trackless-trolley operator, particularly if he is inexperienced, tends to pull to the curb and take a turn at too sharp an angle; then, if contact with the trolley is lost, the cars become dark. An automatic switch controls the emergency system which gives head-lamp and tail-lamp illumination and four inside or corner lights.

After competitive bidding, the car illustrated in Figs. 24 to 26 was built by the Trackless Transportation Co., in cooperation with the Brockway Motor Truck Co., the Osgood Bradley Car Co. and the General Electric Co., under the supervision of inspectors of the Department of Plant and Structures. This general design has now been adopted as standard for the City's trackless-buses. Fifteen were ordered, 8 being designated for the Staten Island Richmond-Tottenville line and 7 for City Island. This new Richmond-Tottenville line is supplied with power from a City plant at Seaview Hospital by a 400-kw. 6600-volt General Electric generating-set, the power being transmitted about 3 miles by a three-phase transmission-line to a substation at Greenridge on the Arthur Kills Road over which the buses operated. This substation is a 300-kw. 600-volt automatic-substation constructed by the Westinghouse company. The operation of this line during the winter and up to the present has been most satisfactory.

OPERATING COSTS

Commissioner Whalen has recently made a survey of the operation, maintenance, depreciation, administration and interest charges on the trackless-trolley systems operated on Staten Island to determine whether this type of service over a total of approximately 16 miles of road or from 28 to 30 miles of line is cheaper than that obtainable by gasoline buses operating with the same schedule and with the same number of cars. It was found that, taking all of the above items into account, the trackless trolley could be operated at a cost of 25.6 cents per car mile; whereas, the gasoline-bus operation would have cost 33.9 cents per car mile.

If the trackless-trolley buses were increased in number with a corresponding increase in the gasoline buses, the difference would be even greater, because the overhead construction which costs about $6,500 per mile of route becomes a smaller and smaller factor in the computation of the fixed costs.

THE "PERMANENCY" FACTOR

Trackless-trolley systems have a value over motorbus lines for undeveloped-locality transportation which cannot be directly expressed in dollars and cents per car mile, though it is made very apparent in "realty-tax" returns. This is the effect of the idea of *permanency*

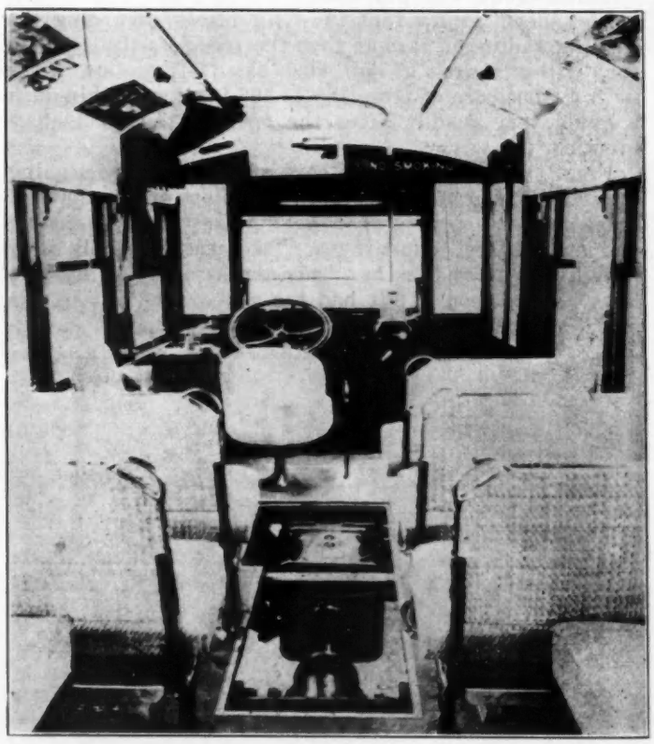


FIG. 25—INTERIOR VIEW LOOKING FORWARD IN ONE OF THE CARS ILLUSTRATED IN FIG. 24

The Accessibility of the Motor by Removing a Trap Door Should Be Noticed; also the Positions of the Motor Controller, the Steering-Wheel and the Registering Fare Box

resident in the trackless-trolley overhead-construction. The valuation of real estate along the Staten Island trackless-trolley lines has increased enormously. That the psychological effect obtains can be emphasized further by calling attention to the experiences of the United Railways & Electric Co., at Baltimore. This corporation operated a 7-mile motorbus-line in the outskirts of the city and gave excellent service; but, as no building activities followed, the line was abandoned. Later, at the request of the Public Service Commission of Maryland, the company constructed a trackless-trolley line over the same route and, within 3 months, marked building activity commenced. Eighty homes were erected and real estate increased about 20 per cent in value.

Trackless Trolley Advantages

We have found that the trackless trolley accelerates more rapidly, more uniformly and with greater comfort to the passengers than the gasoline bus can possibly do. We find it takes the hills, which are very pronounced on the Tottenville line, with ease.

The advantages of the trackless trolley over those of the gasoline-driven bus are

(1) Lower operation expenses
(2) Unlimited heat
(3) Unlimited light
(4) Less noise when starting or accelerating
(5) No gas fumes
(6) Better hill-climbing power

I feel that Commissioner Whalen has shown considerable vision in his selection of the trackless trolley as the means of transportation in the lightly populated sections of Greater New York City; and further, that it must be very seriously considered by all railway officials who wish to extend their systems without encumbering themselves with both excessive fixed and operating costs.

Ethelbert Favary:—The advantages of the gasoline motorbus have scarcely been mentioned and the advantages of the trackless-trolleys and electrically operated cars have been overemphasized. The trackless trolley may be more economical and more efficient in outlying districts, but it is impractical for city use. I also disagree with Mr. Kennedy's statements that

> Street-railway organizations ..... are qualified ..... to render public service, modified wherever necessary to meet changing demands
>
> Ultimately, public transportation must certainly remain in their hands
>
> The vehicles which would best serve these purposes would be those that could utilize partly, if not entirely, the power supply and distribution system available ..... existing within the street-railway organizations. By such means existing properties representing large investment can be preserved.....

These views may do incalculable harm to motorbus transportation-systems. For instance, the suggestion has been made that the street-railway companies in New York City receive franchises to run motorbuses along the different avenues instead of their surface lines, since the traffic could be increased one-third by such a change. The surface cars, being in the middle of the street, delay all fast-moving traffic, while motorbuses can receive or discharge passengers at the curb. Many other advantages are provided by motorbuses, and many disadvantages are inherent in electric surface-lines, but greater comfort and speed are furnished by the motorbus.

Motorbus Lines in New York City

The capitalization of the street-railway companies is more than $500,000 per mile. According to Commissioner Whalen, the bus lines now in operation on New York City streets can be equipped at an average cost of $22,800 per mile, including the motorbuses and garage equipment. Even with this figure doubled, it would still be less than one-tenth of the outlay of the street-railway companies per mile of line. Six per cent on $500,000 is $30,000; this is the amount the lines would have to earn per mile, in addition to their running expenses, to pay the interest charges alone.

To run a motorbus line with such a heavy initial-handicap means either inferior service or an increased fare, or both, to make the line profitable. The interest charges per mile of street railway alone is sufficient to renew the entire bus equipment every year; hence, a new company relieved of that heavy burden to begin with, could

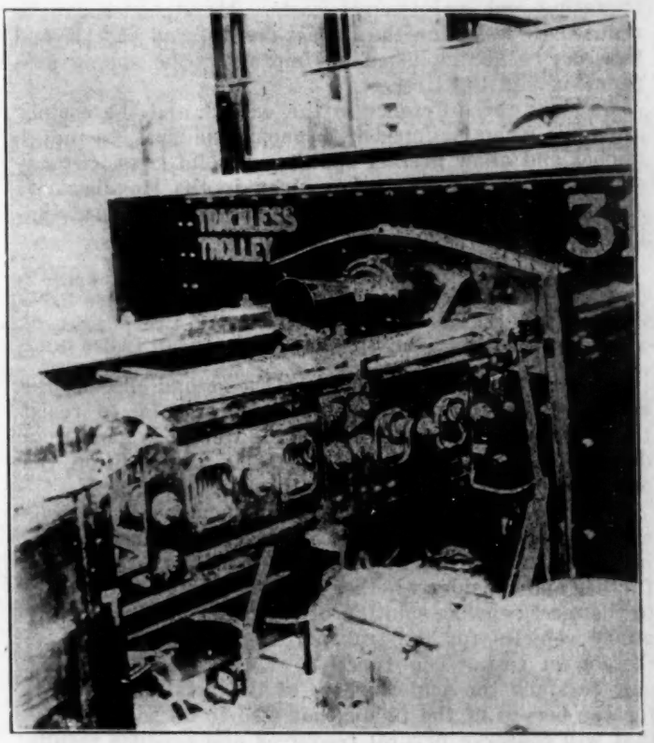


Fig. 26—How the Control Resistances, the Storage-Battery and the Emergency Lighting-Generator Are Placed under the Hood at the Front of the Vehicle

give better service at less cost. No doubt these conditions obtain in many other cities and, while it is advantageous for the street-railway companies to run trackless-trolleys in outlying districts, they may make failures even of these, due to their over-capitalization.

The argument has been advanced that, on many lines, the motorbus does not give such an assurance of permanent service as the electric street-car lines and therefore, property values do not increase as they do along street-car routes. Examples are cited in which motorbus lines have been discontinued and in which many others are losing money. Why cannot motorbus companies be as responsible as the street-railway companies? As soon as our cities are prepared to give motorbus companies regular franchises over good routes, such as New York City avenues, these companies will be as responsible as the street-railway companies, if not more so, and will not be losing money. Many motorbus lines failed because the street-railway companies had the franchises over all the good lines, while motorbuses only had franchises over

those lines where the traffic was slight or where the street-railway companies had abandoned service, due to the paucity of traffic. As soon as motorbus companies are given proper franchises, they will become responsible and their service can be made just as efficient as that given by the Fifth Avenue Coach Co.

Motorbus Economy

From Mr. Kennedy's figures, the operating expense of the trackless trolley is less than that of the motor omnibus when the interest rate from capitalization is not considered. While this may be true today, the motorbus is a comparatively new product and, due to improvements that are being made continuously, we can expect a more efficient utilization of fuel and a reduction in the operating and maintenance costs. Maintenance can be reduced by lessening the impact due to road shocks, and this can be accomplished by improving the spring suspensions and the tires.

To show the difference in the weight and the cost of a chassis when equipped with pneumatic tires, to reduce impact and allow greater speed, and solid tires, two examples of well-known makes given in the Handbook of Automobiles, published yearly by the National Automobile Chamber of Commerce, are

1½-ton chassis equipped with pneumatic tires; price $1,097; chassis weight, 2390 lb.; gross weight, 6080 lb.
1½-ton chassis equipped with solid tires; price, $3,450; chassis weight, 5700 lb.; gross weight, 12,000 lb.

Because solid tires are used on the second chassis, the entire construction must be more substantial to resist the greater road impacts, the chassis weight is more than double and the cost is more than treble. In addition, the operating costs are necessarily higher.

Joseph A. Anglada:—Certain fundamental reasons should cause the automotive engineer to pause in his intensive efforts to improve the gasoline powerplant further and consider whether he is directing his energies to the most sensible solution of the problem of providing public vehicles for the transportation of passengers at the lowest cost. The function of the engineer is not only to apply the achievements of the arts and sciences for the benefit of the public, but also to do so in a way that will conserve natural resources and produce results at the minimum cost. He should not merely provide a technical solution of the problem but a commercial one. This raises the question whether, in the field of the motorbus, the horizons of the engineer and the business man have not been clouded sufficiently by their close familiarity with the internal-combustion-engine type of vehicle to cause them to lose sight of the applicability of the electric vehicle to this arduous service.

If the broad unbiased mental outlook is preserved, the characteristic features of lower operating cost, long life, low maintenance cost, simplicity, cleanliness and safety of the electric vehicle are recognized as being essential to providing the best motorbus service from the standpoint of the public and the operators of the service. It may be argued that someone at some time has tried electric vehicles in this service without success, but an investigation of the causes will show that, under the same conditions, any other type of vehicle would not have succeeded.

Severity of Motorbus Service

A survey of the conditions under which motorbuses operate in urban districts reveals a close similarity between electric-street-car service and motorbus operating conditions. The frequent stopping and starting of the vehicle, as often as 15 times per mile traveled and averaging 6 times per mile, imposes a service on the propelling and speed-controlling machinery of the vehicle that exceeds in severity that due to continued high-speed operation on good roads. Compared to the electric vehicle, the gasoline-engine vehicle is not, nor in the light of present knowledge can it be, adapted to withstand the high duty of motorbus service as efficiently as the simpler and more reliable electric vehicle. For proof, it is necessary to consider the few gasoline-motorbus installations extant that have made money for their operators over a period of years and to ascertain the reason. Analysis will reveal that, for the first year or so, the gasoline equipment operated satisfactorily because it was new. After the second year, the depreciation on the equipment, due to the nature of the service and the high-speed performances indulged in by the drivers, increased greatly, while the necessity for adjustments and repairs became annoyingly important. After a period of from 3 to 5 years, the maintenance charges became so great as to prohibit the safe use of the vehicles for public service by responsible organizations. The result was that the so-called outlaw or jitney operator entered the field and the service suffered still further.

Operating electric street-cars in the dangerous physical condition existing in many motorbuses would not be tolerated by the public or the operating companies, and when it is known that the electric street-cars are from three to five times as old as the gasoline vehicles and have cost decidedly less to maintain during their life, it will be conceded that electric propulsion of passenger-carrying vehicles operating on tracks in city service possesses fundamental advantages that can be applied to vehicles operating on rubber tires.

Long life, low operating and maintenance costs and ease of control are the fundamental advantages of electric vehicles, whether of the storage-battery, trackless-trolley or self-charging type. In addition, flexibility of the chassis construction permits great passenger accommodation with a low floor-level in a simple inexpensive chassis having a short wheelbase and consequently a short turning radius; also, with the body overhang beyond the axles reduced to sensible proportions.

Arthur J. Slade:—Supporting Mr. Kennedy's contention for an analysis of passenger-transportation problems in given instances and the production of suitable equipment to meet the specific requirements of each installation, some European experience is cited.

In France, the Paris, Lyons & Marseilles Railway has provided passenger buses and touring coaches by which tourists may continue their excursions beyond the rail terminals; for example, through the French Alps and, more recently, along the northern coast of Africa through the French colonial possessions. These extensions not only produce revenue in themselves, but bring added passenger traffic to the railroad. In Paris, the subways, the trolley lines and the buses are operated by a common transportation company under municipal supervision. The subways run to most of the old-city gates; the trolleys operate largely outside the city proper, serving the suburbs but running into convenient central locations within the city. In the most congested districts the buses, operating on the zone-fare system, serve the short-haul traffic, the routes being laid out, changed or extended to meet the requirements of all localities from both the viewpoint of revenue and that of public service. The three mediums of transportation are run in co-operation, not in competition. At present, trolley-bus

extensions are being installed which may eventually become trolley railway-lines, as the suburban growth of Paris is increasing very rapidly. Gasoline motorbus routes are being added and extended as the area occupied by the old-city fortifications, which are being razed, becomes occupied by apartment houses and dwellings. The passenger-transportation problem is being handled broadly and comprehensively, without prejudice in favor of one medium over another. In certain parts of Paris the traffic congestion has attained serious proportions and, in the summer of 1922, the subject was discussed at some length in the newspapers. In this connection, some of the details of the latest Paris buses, as described in a recent issue of a leading French engineering publication,[27] are interesting.

The Six-Wheel Motorbus

Regarding the six-wheel motorbus, this publication suggested, as far back as 1904, means whereby heavy-duty vehicles could be made to carry heavier loads on six wheels than on four. This idea has been taken up by the transportation company of Paris with the purpose of increasing the carrying capacity of the motorbuses that operated in the center of the city, the increase in the congestion of passenger transportation having led to this study.

On the trolley lines, cars hauling trailers offer a combination seating 100 passengers and even more at peak loads. The motorbus, which is particularly popular, has only been able to carry 38 passengers. While it would be possible to mount an upper deck to increase this capacity, as was the former practice with the horse-drawn omnibuses and the earlier motorbuses, and notwithstanding the custom in England, where the double-deck motorbuses and trolley cars are generally used, the double-deck arrangement has been definitely discarded in Paris for the omnibuses as well as for the trolley cars, principally on account of the delay that this arrangement requires for the entrance and the departure of passengers, certain trolley-car lines in the suburbs being the only exceptions.

Until recently, the omnibuses with a capacity of 38 passengers have been satisfactory, but the necessity of increasing the capacity has induced the study of a solution involving an increase in the body length to be carried by three axles. One of these buses with six wheels and a capacity for 48 passengers has already been placed in experimental service on the line from the Madeleine to the Bastille, which follows the grand boulevards, and an equipment of 50 more such vehicles is now under construction.

The new motorbus has a total length of approximately 34 ft. and an overall width of approximately 7½ ft. The three axles are unequally spaced. The forward wheelbase is approximately 14 ft. and the rear wheelbase approximately 7 ft. The front and the rear wheels are steering wheels, those in the center of the vehicle not having steering functions. All of the wheels are 38 in. in diameter, the forward and the rear wheels having single tires and the intermediate wheels dual tires.

In the case of a vehicle having two axles, the distribution of the load on the wheels is automatic regardless of road irregularities, which is not possible in a six-wheel chassis without recourse to a special construction, consisting of what may be described as a balancing lever joining the forward ends of the rear springs to the rear ends of the intermediate springs. This balancing lever pivots on an axis carried by a bracket attached to the chassis. With this arrangement, the applied weight of the chassis on the springs is distributed automatically, regardless of the relative positions of the wheels themselves.

By a predetermined location on the center axle, the weight on the driving wheels can be proportioned, which is a valuable consideration in the improvement of the tractive effect of the driving wheels. An added advantage of this balanced suspension is to reduce the vertical displacement of the chassis, reducing shocks and tending to diminish the wear on both the vehicle and the pavement. Two brakes, one on the transmission shaft and the other on the intermediate or the driving wheels, are provided. The engine of the new motorbus is of the same type as that employed on the four-wheel buses of the company, and the other general features of the powerplant are similar to those previously in use.

The forward and the rear wheels are steering wheels, and these four wheels, which are mounted on conventional steering-pivot axles, are connected together with lever arms and rods to turn, in steering, so that their radii converge at a central point that is also on the axis of the intermediate driving-wheels. Inasmuch as the rear steering-wheels, to accomplish this result, do not turn to as great an angularity as the front steering-wheels, a difference of leverage is introduced in the steering connections. The six-wheel vehicle has less tendency to wabble or depart from the straight line than the four-wheel vehicle, and greater steering stability results.

The article in *Le Genie Civil*[27] is illustrated by views showing the overall dimensions, the steering linkages, the spring-suspensions and the turning radius, and it is interesting to note that the clearance circle for the front end of the vehicle has a radius of approximately 28 ft. and the rear end of the vehicle 27 ft., while the turning radius of the inner of the driving wheels is approximately 16½ ft. When empty the total weight of the omnibus is approximately 16,000 lb., which is a little more than 300 lb. per passenger or very much less than that per passenger on the trolley cars, which is nearly double. On the latest trolley cars, the weight per passenger has been reduced to slightly more than 500 lb. Nevertheless, it is hoped to reduce the six-wheel-omnibus weight on a new model that is now being designed for either a gasoline-engine powerplant or an electric motor with a double trolley to about 260 lb. per passenger.

Except for its greater length, the body construction is arranged similarly to that of the former type of bus. The forward or first-class compartment seats 20 passengers instead of 16, and the second-class compartment seats 20 instead of 12. On the other hand, the platform carries only 8 standing passengers instead of 10. This reduction in the platform capacity is due to the narrow shape that has been given to the platform to increase the clearance on turns. The new bus, therefore, carries 48 instead of 38 passengers. In all the Paris buses the seats are arranged so that about an equal number face forward and to the rear. Standing passengers are not permitted inside, but only on the platform. Largely increased carrying capacity would be possible if the aisle space within the body might be occupied by standees.

Finally, a further improvement recently proposed is the replacement of the solid by pneumatic tires, which will unquestionably increase the comfortable riding qualities of the vehicles. An omnibus equipped with pneumatic tires has been put on trial, and the results obtained by the tests will decide whether it is advantageous to adopt pneumatic-tire equipment. The foregoing installa-

[27] See *Le Genie Civil*, Dec. 30, 1922, p. 605.

tions serve to emphasize the argument presented in Mr. Kennedy's paper, in favor of coordination of railway and automotive engineering effort.

W. P. KENNEDY:—Mr. Fielder mentioned reasons why the gasoline-electric buses on Fifth Avenue were not continued. I wish to point out that those gasoline-electric buses were built at a time when there were few gas engines to choose from. In fact, the sources of supply were so limited that special engines had to be built for them. These engines were not at all as satisfactory as they might have been and were afterward superseded by De Dion engines.

About the possibilities of the gasoline-electric bus and its being continued, I quote part of a letter dated April 17, 1919, from R. M. Meade, then president of the Fifth Avenue Coach Co., in which he covers this subject as I have stated it.

> Confirming what I said to you today, if the engines of the gasoline-electric buses had been as good as the De Dion engines, we would not have gone abroad for the 25 chassis we recently ordered.
>
> This statement is made in the light of the results obtained from the experimental installation of the De Dion engine in bus No. 156, as the performance of that bus shows that the electric-transmission feature is a complete success. At the time we ordered the 25 chassis referred to, we had not had the benefit of that experiment.

This indicates clearly that if he could have got gasoline-electric buses having proper engine equipment at that time, he would have continued their use.

Mr. Horner pointed out that no progress has been made in trolley-bus installations in England in recent years. The conditions of operation over there are not such as we have to contend with, in the changing conditions here at present. The tendency to put out new vehicles is indicated in what the Associated Equipment Co. has done in developing its new-type trolley-bus equipped with a single motor. The conditions with which the street railways have to contend at present in this Country make the situation entirely different from the one that has existed in England; consequently, under these conditions, an opportunity exists to develop the trolley bus much more effectively than the opportunties in England afford.

Another point questioned had reference to the feasibility of the street railways rendering service in the transfer of freight and particularly such services as is exemplified by the store-door delivery, whether such service would produce congestion of traffic or whether a good business opportunity warranted this or not. My recommendation in this direction was to show the possibility that exists for the street railways if they want to go into the freight-handling business. In making that recommendation I naturally do not have in mind such cities as New York or the larger cities where perhaps the street railways would not be justified in entering into that phase of business. It appears that in the numerous minor cities throughout the Country, such as Dayton, Ohio; Rochester, N. Y.; Columbus, Ohio, and dozens of other such places where store-door delivery has not the conditions of congestion that exist in New York City, it would be feasible for street railways to go into that business for profit.

Mr. McPherson asked why storage-battery equipments were not recommended. Nobody is a stronger advocate of the use of storage-batteries than I have been, because I have been responsible for putting more of this type into use than many other engineers have done. I am in favor of the storage-battery, but it is a hard job to sell it. Nobody wants to take up the burden of it. Even the storage-battery companies do not, and neither do the vehicle builders; therefore, I do not feel justified in shouldering such a large proposition myself. But I firmly believe that the electric-storage-battery vehicle, rationally applied, would be the most economical one that a street-railway company could use, especially when it has cheap power, and this is substantiated by what Mr. Anglada has said.

BUSINESS FORECASTING SYSTEMS

IN attempting to forecast the trend of business, two fundamental principles are involved. The so-called modern forecasting system aims to make reliable estimates of the future by applying the theory of causation. Predictions are based upon the assumption that, since economic phenomena are responsive to fundamental laws, therefore approximately similar causes should produce similar results, interpretations of possible results being based upon past experience. This method has much to commend it; and its value is of course dependent, in a large measure, upon the accuracy of the statistical material used and the degree to which similar material of the past is available as a basis for making real comparisons. As our information and statistics become more accurate, and as more experience in interpreting them is acquired, the greater will be the value of studies of this nature.

The stock-market method, on the other hand, follows a different principle, which has also many features to commend it. In theory, it aims to predict, by reflecting in a definite manner, the dominant opinion upon which business men are acting after important decisions have been made. Thus, to the extent that current business opinion makes a proper estimate of the future the market should tend to anticipate conditions.

Both methods of attempting to predict the future of business appear to have merit; neither, however, has achieved for itself a place that makes it infallible. The factors influencing the trend of business conditions are as complicated and difficult to appraise as human nature itself. Economic factors are responsive not only to certain basic, physical and so-called economic laws but are also modified to a great extent by the injection of psychological factors of which we as yet have little understanding. For this reason great care and judgment must be exercised in dealing with economic data that aim to forecast the future. No "rule of thumb" method has yet been found that can be substituted for individual research, judgment and hard-headed business acumen.—H. F. Boettler, First National Bank, St. Louis.

Illumination Diagrams for Showing Head-Lighting Device Performance

By F. H. Ford[1]

Mid-West Section Paper *Illustrated with* Photographs and Charts

INSTEAD of representing light intensity by lines to indicate photometric values the author recommends an arrangement for denoting the intensity by varying degrees of tint on the surface of a chart that is supposed to represent the roadway. In the opening paragraph the thought is brought out that present-day automobile lighting equipment is not designed in such a way as to make its performance a selling feature and the several reasons why the efficient distribution of light on the road has been overlooked are pointed out, emphasis being laid on the fact that the average car-designer is not an illuminating engineer, and that even if he did wish to use the best light available on the car he would have to make personal tests of the devices under all conditions of night driving before being in a position to recommend the most efficient head-lighting device.

The method of comparing devices by illumination charts that illustrate the performance of head-lighting devices is described, and charts from four typical specimens chosen from the many that are on the market are reproduced to show that while the light from each of the devices does not exceed the glare limit above the level of the head-lamp and is at least sufficient to meet the minimum driving-light that the law demands below the level of the head-lamp, a marked difference in the efficiency of these equally legal devices will be found. The construction of the devices for which the road illumination charts are reproduced is described.

Tabular data show the report of a test of the candle-power distribution from the combined beams of a pair of standard head-lamps as directed toward various points on a vertical screen located 100 ft. ahead of the test lamps. The outlining of areas of equal illumination is described at some length, the text being supplemented by diagrams and tables.

THE lighting equipment of present-day automobiles is not designed in a way that makes its performance a selling feature. True, various attempts have been made at the modification of the cone-shaped beam of light that comes from an electric bulb within a parabolic reflector by glass fronts or lenses, and there have been attempts to use specially shaped reflectors that deviate from the true parabola. These modifications, however, have been largely made with a view to eliminating glare, and if the thought of improving the distribution of the light on the road has been present, it has, judging from the results obtained, been a secondary consideration.

Efficient distribution of light on the road has been overlooked for several reasons. The average car designer is not an illuminating engineer. He does not design his own head-lamps. If he has a hobby, it is more likely to run to some item of mechanical equipment or to body design. Head-lamps are to him a necessary evil, like a kit of tools or a horn or spark-plugs, something to be bought as cheaply as possible and consequently within the jurisdiction of the purchasing, rather than the engineering, department. What if the light is not so good; cars do not often have their driving lights compared. If the lamps look good on the outside and if they look sufficiently legal so that dealers will not enter a protest, nothing more is necessary.

Furthermore, even if the car designer realized that exceptionally good driving-light might enhance the value of his car, and if he had made up his mind that he wanted the best light available, he would have a hard time making a choice without personally testing dozens of devices under all conditions of night driving. He can easily eliminate those that are illegal by referring to the lists published by the various States. From these lists he could even determine which device was the most efficient from the standpoint of stopping glare. But nothing has been published to show the comparative lighting efficiency of head-lighting devices.

If it were a case of lighting a factory or an office, no trouble would be experienced in choosing proper equipment by comparing the illumination charts that are available for industrial lighting fixtures. But despite the importance of good driving-light to the comfort and safety of the driver, no data more useful than the claims of manufacturers have been published.

Use of Illumination Charts

In this paper we shall show a method of comparing devices by illumination charts that illustrate the performance of several typical specimens chosen from the many that are on the market. It should be noted that the light from each of these devices is perfectly legal; that is, it does not exceed the glare limit above the level of the head-lamps and it is at least sufficient to meet the minimum driving-light that the law demands below the level of the head-lamps. The difference in efficiency between these equally legal devices will be apparent at a glance. The devices used to illustrate this method of comparing the efficiency of driving-lights were selected because they represent the four types of head-lamp equipment most commonly used as standard by car builders.

Devices Nos. 1 and 2 are both lenses designed for use in a head-lamp in which the lamp filament is set at the point in the reflector that will give a beam of the smallest possible diameter. With this adjustment of the filament and without the lens in place the beam will diverge slightly from the true cylindrical form, because it is compounded of diverging rays from that part of the filament behind the reflector focus, of nearly parallel rays from that part of the filament near the focus and of rays from the part of the filament ahead of the focus that first cross over and then diverge.

With an unmodified beam of this sort, the light would

[1] A.S.A.E.—Engineer and manager of head-light lens department, C. A. Shaler Co., Waupun, Wis.

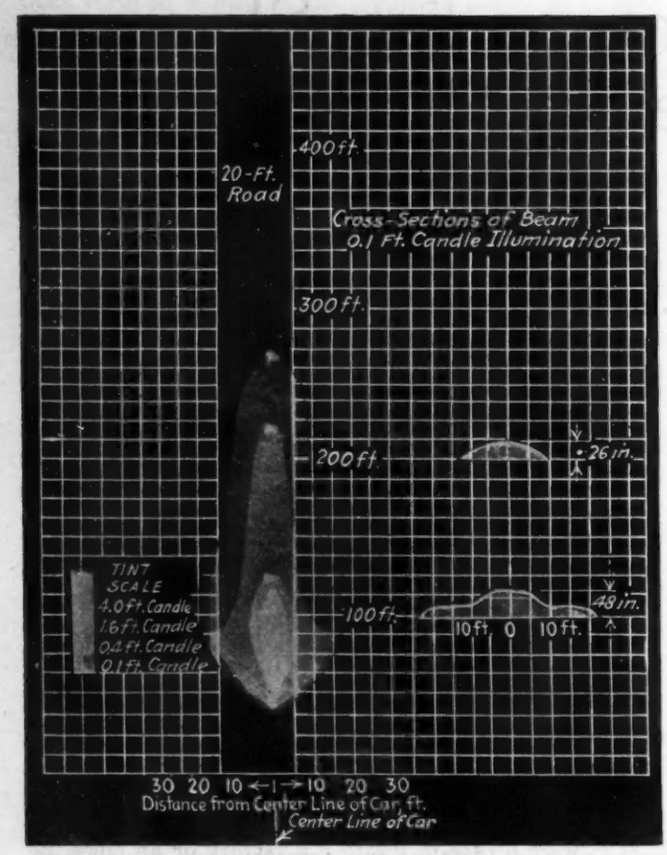


FIG. 1—DIAGRAM SHOWING THE ILLUMINATION AT THE ROAD LEVEL FROM A DEVICE THAT IS DESIGNED FOR USE WITH THE LAMP FILAMENT AT THE REFLECTOR FOCUS AND NO TILT TO THE HEAD-LAMP

be projected to a great distance. It is the searchlight principle, excellent for spotting an object far away, but unsuitable for lighting a road, because the beam is too narrow and would leave the sides of the road and the immediate foreground in darkness. Furthermore, this beam, unless tilted down very sharply, would give an objectionable glare, and then its value for distance use would be lost.

Device No. 1 bends the beam downward by horizontal prisms that cover the entire opening in the head-lamp. These prisms are proportioned so that, in this particular device, the most intense part of the beam is deflected downward until it strikes the road about 50 ft. ahead of the car when the head-lamp itself is pointed so that its axis is parallel to the road. The width of the lighted area on the road is increased at the expense of its length by vertical cylinders, because the lens that spreads the light sidewise gives the beam cross-section the form of a modified ellipse.

The illumination diagram for device No. 1, reproduced in Fig. 1, shows that the lighted area is widest at a distance of about 100 ft. in front of the car and narrows rapidly beyond that distance. The maximum range of the 0.1-ft-candle illumination is only about 250 ft. The diagram showing the cross-section of the beam also shows that the depth of the light on the road decreases rapidly as the distance from the car increases; in other words, that the top of the beam slopes downward so that at 250 ft. ahead it would be difficult to detect an object suspended 1 ft. above the road. This extreme slope of the beam is made necessary by the fact that the unmodified beam from the reflector is a compound beam, composed of divergent and convergent as well as parallel rays, and to eliminate the glare of the former, all the rays are bent downward, as a group. To separate the different kinds of ray is impossible, and all must be deflected downward together.

In device No. 2, for which the illumination diagram is reproduced in Fig. 2, no provision has been made within the lens itself for bending the beam downward to comply with the glare limits set by Illuminating Engineering Society specifications. This must be done by tilting the head-lamp itself, a thing that is not often done because of its unfortunate effect on the driving light.

Devices of this type are provided with only vertical cylinders, which spread the light laterally and, in general, spread it wider than do devices of the design of No. 1. This greater lateral spread has the effect of making the modified beam into a comparatively thin sheet of light. Aside from the greater width of the lighted area, the distribution of light from device No. 1 is very much like that of No. 2. The slight difference shown in the diagram is due largely to the fact that the prisms of the No. 1 device gave the beam a little more tilt than was necessary to avoid glare, while the beam from No. 2 was tilted just enough to make it meet the Illuminating Engineering Society's requirements.

With both types of device the brightest light is near the axis of the beam, and when the beam is directed downward sharply enough to comply with the glare limits it causes an over-illumination of the road about 50 ft. ahead of the car, directly in the driver's line of vision.

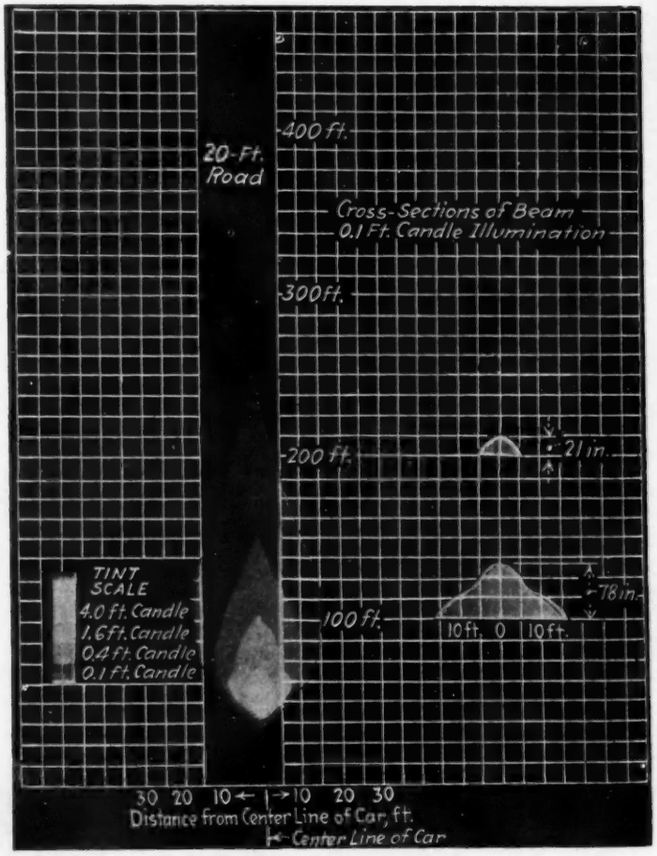


FIG. 2—ILLUMINATION AT THE ROAD LEVEL FROM A DEVICE THAT IS DESIGNED FOR USE WITH THE LAMP FILAMENT AT THE REFLECTOR FOCUS AND WITH THE HEAD-LAMP AXIS TILTED 12 IN. AT 25 FT.

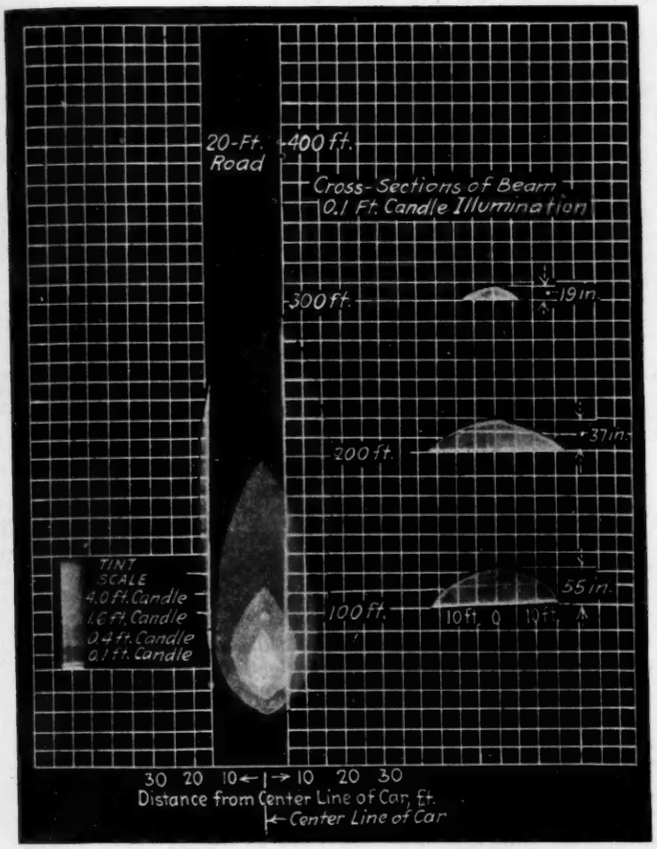


FIG. 3—A DEVICE MADE UP WITH A SPECIAL REFLECTOR AND A PLAIN COVER GLASS ILLUMINATED THE ROAD AS SHOWN IN THIS DIAGRAM

Device No. 3, the illumination diagram of which is reproduced in Fig. 3, illustrates one type of specially formed reflector that throws a beam having an elliptical cross-section, instead of a round beam, as does the parabolic reflector. The light is distributed in practically the same manner as with lenses such as No. 1 and No. 2, except that the beam is a little brighter in the center. The total width is somewhat greater than can successfully be obtained by using vertical cylinders to spread out the highly concentrated and nearly cylindrical beam from a standard parabolic reflector. The great accuracy required in forming the surfaces of a reflector of this type and in locating the filament within it seem to present a problem in manufacturing it in commercial quantity.

Device No. 6 is a type of lens designed for use with a head-lamp having an adjustable filament back of the focal point of a regular parabolic reflector. When the filament is in this position all the rays will diverge from the axis of the beam. The intensity is greatest near the center of the beam and decreases toward the outside. The unmodified beam is a fairly wide-spreading cone with the brightest part near the center. All the light from the lower half of the reflector is directed below the level of the reflector by the reflector itself. As it is unnecessary to tilt this part of the beam to avoid glare, the beam can be used to full advantage for lighting distant parts of the road. The brilliant part of the beam is allowed to shoot forward parallel with the road instead of being aimed downward.

Since the most intense part of the light in the lower half of the beam is projected at the smallest angle below the level of the head-lamp, the top of this part of the beam is the brightest, and this bright part intercepts the road farther ahead of the car than do the weaker parts of the beam. The less intense light, which is projected more sharply downward by the reflector, intercepts the roadway nearer to the car. By means of vertical prisms or cylinders the lateral spread of this part of the beam can be increased or decreased as may be desired. The light from the upper half of the reflector can be redirected below the level of the head-lamp to avoid glare and to give additional illumination at points of the road that are not sufficiently illuminated by light from the lower half of the reflector.

The illumination diagram for device No. 6 is reproduced in Fig. 4 and shows the effective distribution of light that can be obtained by redirecting the light from a divergent beam by suitable prismatic elements. In this particular device the light from the lower half of the reflector is used to furnish the illumination straight ahead in the path of the car. The light from the upper half of the reflector is divided and is directed toward both sides of the road, furnishing illumination along the edges for which a spotlight would ordinarily be used and adding to the width of the illuminated area for 100 ft. ahead of the car. This additional illumination is to the side of the driver's direct line of vision so that it does not interfere with his ability to distinguish objects at a distance.

A comparison of the illumination diagram for the four devices shows that no inherent property is possessed by either lenses or reflectors which makes either group the only solution of the problem of securing good driving-

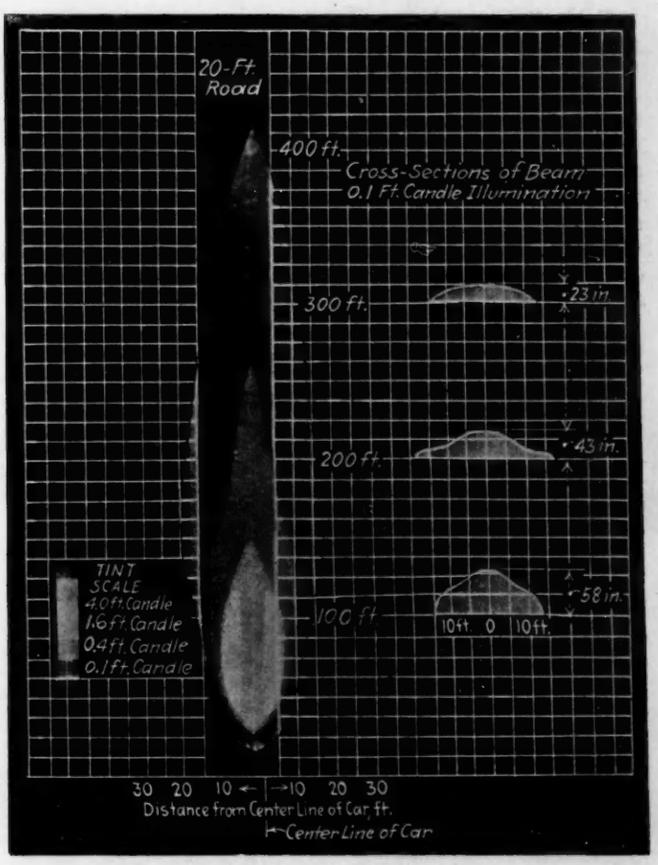


FIG. 4—DIAGRAM SHOWING THE ILLUMINATION AT THE ROAD LEVEL FROM A DEVICE THAT IS DESIGNED FOR USE WITH THE LAMP FILAMENT BACK OF THE REFLECTOR FOCUS

TABLE 1—CANDLEPOWER DISTRIBUTION AS AT 100 FT. FOR DEVICE NO. 1

Test No. 2—Candlepower Distribution Distance to Left of Median Vertical Plane (Car Axis), Deg.	0	1	2	3	4	5	6	8
Degrees Above Horizontal								
1	1,680	1,600	1,160	710	600	560	480	250
½	2,200	1,900	1,210	....	700	....	....	280
Horizontal	2,900	2,500	1,600	1,140	960	760	630	310
Degrees Below Horizontal								
½	4,500	3,900	2,400	1,520	1,200	1,010	900	480
1	9,000	7,600	3,900	2,600	2,200	1,800	1,470	670
1½	18,000	16,200	9,300	5,800	4,100	3,500	2,700	1,130
2	46,000	36,000	21,000	10,700	9,200	7,400	6,000	2,500
2½	78,000	62,000	34,000	19,600	16,600	....	....	5,200
3	90,000	68,000	38,000	23,000	17,600	16,500	13,500	6,000
3½	62,000	47,000	25,000	....	12,800	....	....	4,300
4	29,000	20,000	11,900	8,200	7,000	5,200	4,500	1,880
5	4,100	.....	....	....	1,260	....	....	180

light. The diagrams, of lenses at least, show that the simple convergent, or divergent, beam from a parabolic reflector, having a lamp filament either ahead of or behind the focal point, can be more effectively redistributed than can a compound beam, when the filament is centered at the focal point.

The illumination diagrams for device Nos. 1, 2 and 6 have been derived from tests made by the Electrical Testing Laboratories of New York City. These tests were made in accordance with the 1922 Illuminating Engineering Society specifications for State approval tests. The diagram for device No. 3 was derived from distribution curves published by the manufacturer of the device.

CANDLEPOWER DISTRIBUTION

Table 1 shows the report of a test of candlepower distribution in the combined beam from a pair of standard head-lamps mounted as on a car and equipped with the device under test. The figures in the table show the luminous intensity, expressed in candlepower, directed toward various points on a vertical screen located 100 ft. ahead of the test-lamps. The position of each of the test-points on the screen is defined by giving the angular distance of the point to the right or to the left of a vertical plane passing through the center-line of the car, and by giving the angular distance of the point above or below the level of the center of the head-lamps.

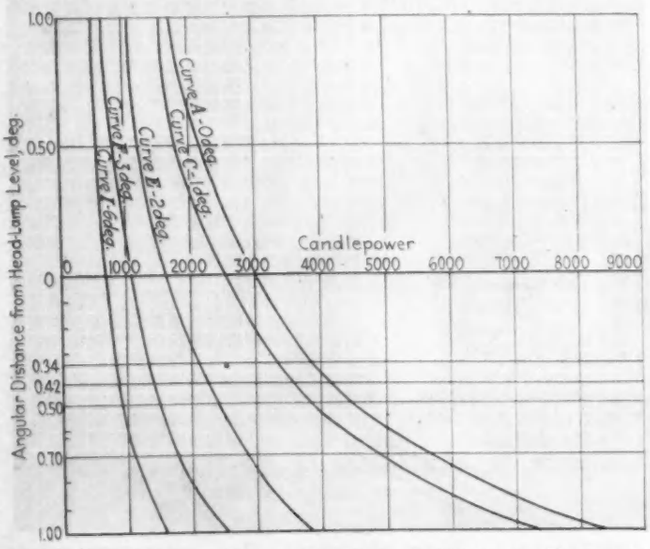


FIG. 5—CANDLEPOWER CURVES FOR VARIOUS ANGLES OF TILT FROM THE HEAD-LAMP LEVEL

The first column shows the angle above or below the level of the center of the head-lamps and the other columns give the candlepower of the beam directed at various angles to the right and left of the median vertical plane between the test-lamps.

To lay-out the outline of areas of equal illumination with a fair degree of accuracy we should have information as to the illumination at the distances ahead of the car that are shown in the second column of Table 2. Assuming that the centers of the head-lamps are 3 ft. above the road level, the first column of Table 2 gives the angular distance below the head-lamp level of a line from the center of the head-lamp to the road level at the distance from the lamp given in the middle column of Table 2. By comparing the first columns of Tables 1 and 2, we find that we shall have to devise some means of securing data as to candlepower for the test-points 0.34, 0.42 and 0.70 deg. below the level of the head-lamps.

These candlepower values can be determined with sufficient accuracy by plotting curves for each of the angles in Table 1. Such curves are shown in Fig. 5. Distances along the *X* axis represent candlepower, and along the *Y* axis, the angular distance above or below the head-lamp level. If we draw a horizontal line at a distance below the *X* axis, corresponding to 0.34 deg., the distances from the *Y* axis to the intersections of this line with each of the curves will represent the candlepower directed toward the test-points 0.34 deg. below the level of the head-lamps. The same procedure for points 0.42 and 0.70 deg. below the level of the center of the head-lamps will give the candlepower for the remaining points needed to complete the calculation.

Before calculating the areas of illumination at road level it will be necessary to determine the illumination

TABLE 2—ANGULAR DISTANCES BELOW THE HEAD-LAMP LEVEL OF A LINE FROM THE CENTER OF THE HEAD-LAMP TO THE ROAD LEVEL AT DIFFERENT DISTANCES AHEAD OF THE CAR

Degrees Below Head-Lamp Level	Distance Ahead of Test-Lamps, Ft.	Equivalent of 1 Deg., Ft.
0.34	500	8.75
0.42	400	7.00
0.50	340	6.00
0.70	250	4.30
1.00	172	3.00
1.50	114	2.00
2.00	86	1.50
2.50	68	1.20
3.00	57	1.00
3.50	50	0.88
4.00	44	0.77

TABLE 3—ILLUMINATION DISTRIBUTION AT ROAD LEVEL, FT-CANDLES

Distance Ahead of Test Lamps, Ft. Distance to Left of Median Vertical Plane, Deg.	0	1	2	3	4	5	6	8
500	0.015	0.013	0.008	0.005	0.004	....	....	....
400	0.025	0.022	0.014	0.009	0.007	....	....	....
340	0.038	0.033	0.020	0.013	0.010	0.008	....	....
250	0.099	0.082	0.049	0.037	0.026	0.021	0.019	0.008
172	0.320	0.257	0.132	0.088	0.075	0.064	0.050	0.025
114	1.380	1.250	0.715	0.446	0.315	0.270	0.206	0.087
86	6.200	4.800	2.820	1.450	1.240	1.000	0.810	0.340
68	16.800	13.400	7.300	4.200	3.560	1.040	0.870	0.420
57	27.700	21.000	11.700	7.100	5.400	5.100	4.150	1.850
44	24.800	18.800	10.000	....	5.100	....	....	1.720

at the road level of the various points of the road at which lines projected from the head-lamps through the various test-points given in Table 1 will intersect the surface of the road. Table 3 gives the result of this determination. The illumination values for the various points are found by dividing the candlepower directed toward each point by the square of the distance of the point from the head-lamps.

The diagrams represent the area ahead of the car. Distances ahead are shown on a scale of 12.5 ft. to each square of the diagram; distances to the side, on a scale of 5 ft. to each square. To avoid complication four limits of illumination have been selected and each classification is represented by a distinctive tint on the diagram.

The area of lowest illumination contains all points at which the illumination is at least 0.1 and less than 0.4 ft-candles; the second, at least 0.4 and less than 1.6 ft-candles; the third, at least 1.6 and less than 4.0 ft-candles; and the fourth, 4.0 or more ft-candles. It will help us to realize what these values of illumination mean in the way of making objects visible if we remember that an illumination of 0.1 ft-candle is about three times the illumination given by a full moon on a clear night; under ordinary conditions it would permit us to discern objects at a distance of about 450 ft.

Areas of Equal Illumination

To determine the boundaries of the various areas of illumination we plot curves such as are shown in Fig. 6. Distances along the *Y* axis represent illumination, in foot-candles. Distances on the *X* axis represent the angular-distances of points from the center-line of the car on which the head-lamps are mounted. Taking the figures in Table 3, we plot curves showing the illumination of points at equal distances ahead of the car and at various distances to the side of the center-line of the car. For instance, curve *A*, Fig. 6, gives the foot-candles at 500 ft. ahead of the car and at 1, 2, 3, 4 deg. to the side of the center-line of the car. Curve *B* covers points 400 ft. ahead; *C*, points 340 ft. ahead; *D*, points 250 ft. ahead; *E*, points 172 ft. ahead; and *F*, points 114 ft. ahead. Curves for the other distances ahead of the car are not shown in Fig. 6.

To determine the boundary of the area having a minimum illumination of 0.1 ft-candle, we draw a horizontal line from the point on the *Y* axis corresponding to 0.1 ft-candle, and note the intersection of this line with the curves *A*, *B*, *C*, and so on. We find that the *A*, *B* and *C* curves are beyond the 0.1 ft-candle range and that the 0.1 ft-candle line intersects the *D* curve at 0 deg. to the side. In other words, the limit of the 0.1-ft-candle illumination will be 250 ft. directly ahead of the car. The 0.1-ft-candle line intersects curve *E* 2.6 deg. to the side, and at 172 ft. ahead of the car corresponds to a distance of 7.8 ft. to the side. The intersection with curve *F* is 7.8 deg. or 15.6 ft. to the side.

Completing the operation for the other curves gives a series of points that will outline the area of 0.1 ft-candle illumination, as shown on the diagram for device No. 1. The same procedure for 0.4, 1.6 and 4.0 ft-candles enables us to outline each of the four illumination areas. The diagram worked out by the method described above gives a graphic representation of the illumination at road level and shows to what extent the road will be lighted. A driving-light to be safe must not only show the surface of the road but must also make it easy for the driver of a car to distinguish objects above the road

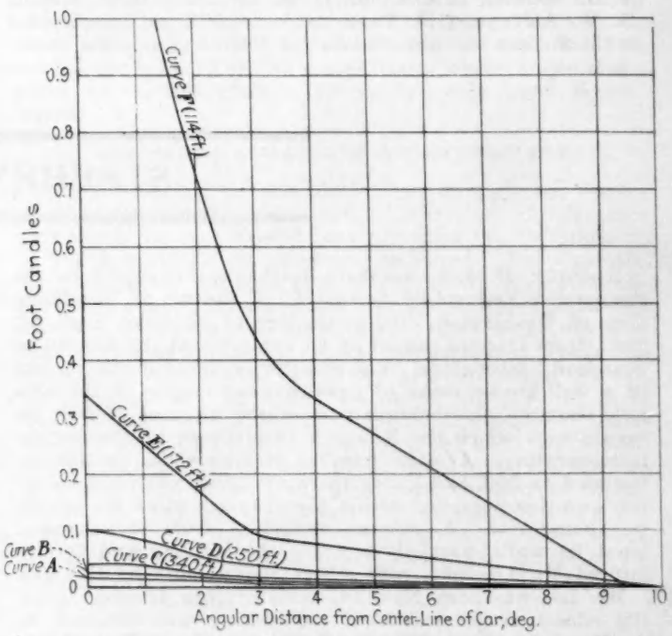


Fig. 6—Curves Showing the Relation between the Illumination in Foot-Candles and the Angular Distance from the Center-Line of the Car

level. By a similar process we are able to determine the beam cross-section at points 100, 200 and 300 ft. ahead of the car. This information together with the diagram of the illumination at the road level will give an excellent idea of the kind of driving-light a given device will produce. Although the diagrams show distinct areas of illumination it should be understood that the actual illumination on the road varies gradually from one area to the next. There will also be illumination beyond the limits of the 0.1-ft-candle area although it is not shown on the diagram.

A. H. WHITING

DEEP regret tinges this recording of the sudden and wholly unexpected death, Sept. 9, 1923, of Allan Hart Whiting, aged 49 years, one of the founder members of the Society and prominent during its early history; and this regret is shared by all who have had the interests of the Society at heart ever since its inception.

In 1904, H. M. Swetland, publisher of *The Automobile*, supported the idea that a society should be formed to benefit all interests of the automotive industry, and tradition has it that discussions held in his office with leading engineers of the industry culminated in a decision to form a society of engineers who were engaged in the design and construction of automobiles. The three men present at this meeting were H. M. Swetland, A. H. Whiting and E. T. Birdsall. An informal organization was effected at that time by naming A. L. Riker as president, E. T. Birdsall as secretary and treasurer, and designating Messrs. Whiting, Birdsall and Swetland a committee to draft a constitution. This committee decided to name the organization the Society of Automobile Engineers, and presented a constitution that stipulated conditions of membership in the Society, specified the annual dues and provided for regular officers and a board of managers.

Under this constitution, in January, 1905, A. L. Riker was elected president; Henry Ford, first vice-president; John Wilkinson, second vice-president; and E. T. Birdsall, secretary-treasurer. Six managers were elected also, namely, A. H. Whiting, L. T. Gibbs, H. M. Swetland, H. P. Maxim, H. W. Alden and H. Vanderbeek. Additional membership in the Society was acquired during 1905, and a formal meeting of the organization was held, Jan. 15, 1906, at the New Grand Hotel, New York City. The meeting comprised a dinner and a program at which engineering papers by Thomas J. Fay, Henry Hess and E. T. Birdsall were presented. At this time the officers and managers mentioned above were re-elected to serve during the following year. Mr. Whiting was present at this meeting.

Mr. Whiting was born at Spuyten Duyvil, N. Y., July 26, 1874. In 1896 he graduated with the degree of Bachelor of Philosophy from the electrical course of the Sheffield Scientific School of Yale University. From September, 1896, to July, 1899, he was connected with the Riker Electric Motor Co., and from that time until November, 1900, was assistant engineer and general superintendent for the Riker Motor Vehicle Co. His later business connections until May, 1905, included engineering, sales and managerial experience in responsible positions with the Electric Vehicle Co., Rainier Motor Corporation, Edison Storage Battery Co. and the Commercial Motor Co. Afterward he became president of the Whiting Motor Co., New York City. For many years this company was a dealer of the Mercer Motors Co., and recently of the Fox Motor Car Co.

Aside from his business, Mr. Whiting's chief interest was in the development of numerous motor-vehicle accessories in his private laboratory. He was a member of the Yale Club, the Engineers' Club and the Automobile Club of America. It is indeed deplorable that he should have become disheartened by a recent swift succession of business reverses that overbalanced his good judgment. He left a widow; there were no children.

STEPHEN M. LEE

STEPHEN M. LEE, assistant mechanical engineer in the Automotive Powerplant Section, U. S. Bureau of Standards, City of Washington, died at the age of 27 years, Sept. 21, 1923, from injuries caused by an explosion at the Bureau of Standard's laboratory. The disaster occurred during a test of a well known make of passenger-car engine in the altitude chamber, this being a part of the program of fuel research with which the Research Department of the Society is cooperating. Aviation gasoline of high volatility was being used as fuel, and low-temperature conditions obtained by refrigeration prevailed within the chamber while the engine was being run. A gaseous explosion inside the chamber burst its walls, partially wrecked the building and fatally injured Mr. Lee who, with others, was conducting the test.

Mr. Lee was born Nov. 14, 1895, at New London, Conn. His education in mechanical engineering was obtained at the Massachusetts Institute of Technology, Cambridge, Mass., and at Leland Stanford University, Cal., during the period 1913 to 1920. From July, 1917, to December, 1918, his practical experience was with the United States Naval Reserve Force. He became a machinist having the rating of first mate on submarine chaser No. 331, and operated its three 220-hp. engines. His subsequent experience prior to 1921 was in operating 10,000 kw. turbines, and in assembling and testing Diesel engines.

Feb. 18, 1921, he became an assistant in the laboratory for aeronautical research at the Bureau of Standards. He gave marked evidence of ability in research work and was promoted to the position of assistant mechanical engineer. His 1923 Annual Meeting paper entitled Testing Fuels for High-Compression Engines, in collaboration with S. W. Sparrow, also associated with the Bureau, and his own later paper for the 1923 Semi-Annual Meeting on Economic Motor-Fuel Volatility, are noteworthy examples of his contributions and indicative of the important work in which he was engaged, for the Bureau, relative to the field of fuel research. He had been assigned by the Bureau to direct the fuel-research work there. His engaging personality, his ability and promise of future achievement, make it most unfortunate that he was not spared to continue his work.

He was elected to Junior Member grade in the Society May 27, 1921.

TENTATIVE STANDARDIZATION WORK

Criticism of all tentative reports should be sent to the Standards Committee in care of the Society

PROPER DIMENSIONING IMPORTANT

Use of Both Bilateral and Unilateral Systems of Expressing Tolerances Proposed

Appreciating the importance of the proper dimensioning and expressing of tolerances in connection with dimensional standards, the Council appointed a special committee early this year to establish some uniform practice. The members appointed to this committee were Earle Buckingham, of Pratt & Whitney Co., G. C. Case, of the Lamson & Sessions Co., and E. H. Ehrman, of the Standard Screw Co., Mr. Buckingham being designated as chairman. Although the committee has not submitted a final report, Mr. Buckingham has submitted a comprehensive review of the situation, together with definite recommendations. It is appreciated that the subject is too far-reaching to expect a satisfactory decision to be reached by a small committee without the general advice of the industry. Mr. Buckingham's article is published herewith to elicit criticism and general information which will be of value to the industry in arriving at acceptable discussions on the problems involved.

TOLERANCES

The subject of tolerances is now being widely discussed, not only here but also abroad, particularly with reference to their application to various standards which are in course of preparation. It has been suggested that some uniform practice in regard to tolerances should be adopted. At the present time, the S. A. E. Standards follow no uniform rule, examples of almost every method existing in them. Before making any definite recommendations, it would be well to consider the various practices to determine if any one method is superior to the others; or, if not, to determine what conditions require different practices.

Tolerances may be divided into two general classes: Unilateral and bilateral. A unilateral tolerance is one wherein the permissible variation is in one direction only from a given nominal size; such as "one inch, plus nothing, minus five thousandths," or "one inch, plus five thousandths, minus nothing."

A bilateral tolerance is one in which the permissible variation is in both directions from a given nominal size; such as "one inch, plus or minus three thousandths."

Another question is often discussed and sometimes confused with the foregoing classification of tolerances, although it is entirely independent of them. This is the question as to whether the shaft or the hole should be standard. For example, if a minimum clearance of five thousandths is required between a one-inch shaft and hole at one time, and a minimum clearance of ten thousandths is required between another hole and shaft of the same nominal diameter, which member should be made of constant size and which should be varied to provide for the differing conditions?

Thus, in order to arrive at a standard practice that will be complete we must decide the following questions:

(1) Shall we use unilateral or bilateral tolerances?

(2) Shall the shaft or hole be basic?

In the second question, the term "shaft" represents all male surfaces, and the term "hole" represents all female surfaces.

One of the first instances in which definite tolerances were established was that of weights and measures used in commerce. In this Country the practice was to specify given tolerances, plus and minus (bilateral), from the fundamental standard. The amount of the tolerances varied according to the use or classification of the standard. The reason for this was to keep the average of the weights and measures used throughout this Country as close as possible to the fundamental units. This practice is shifting, however, as it created a tendency to work as closely as possible to the smallest limit, particularly in the case of standard package goods. For scientific purposes, a bilateral tolerance on weights and measures as small as possible is desirable; while for commercial standards of weights and measures, where the greatest good to the greatest number is desired, a plus tolerance only (unilateral) is gaining favor. While most of the factors governing the practice of tolerances as applied to standards of weights and measures are foreign to those governing mechanical standards, it is of some assistance to review them.

In discussing the subject of tolerances as applied to mechanical standards, we must differentiate between mating and non-mating surfaces. For the first class of surfaces, we must always consider the relationship between the two mating surfaces. For the second class, where ample clearances are always present one surface only need be considered.

NON-MATING SURFACES OR ATMOSPHERIC FITS

For non-mating surfaces, and other unimportant dimensions where tolerances are specified, the class of tolerances that should be used depends entirely upon the requirements of the manufacturing processes employed. For example, the diameter of the head of a rivet, the diameter or thickness of a washer, dimensions of forgings, castings, hot-rolled material, etc., should be permitted to vary in the direction that variations develop in normal manufacture. Thus, if a certain size is controlled by the size of a tool alone, and this tool grows larger with use, the logical thing would be to make the nominal size the minimum limit, and permit a plus tolerance only (unilateral). On the other hand, if, in addition to the wear on the tool, an adjustment of the machine were also involved, so that the normal variation of the dimension in question was either larger or smaller, a plus and minus (bilateral) tolerance should be given. In other words, the type of tolerance used should follow the nature of the variations which exist in the production of the respective material in question.

MATING SURFACES

The problem of tolerances on mating surfaces is entirely different from that of non-fitting ones, or atmospheric fits. We are now considering interchangeable parts. We establish tolerances to assure that under the tightest possible conditions the fit between mating parts will not be too tight; while under the loosest possible conditions they will not be too loose. The first step toward the establishment of tolerances should be to determine the tightest and loosest conditions we are willing to accept. The difference between these conditions represents the sum of the tolerance on the mating parts.

After this value is decided upon the next step is to apportion it between the two mating parts. The way in which this sum is divided should be governed largely by the ease or

difficulty of machining the surfaces in question. Where one member can be held closely to size with relative ease, while its mating member is maintained with difficulty, the more easily made part should have the smaller tolerance. When both parts are about equal in this respect, the tolerance should be divided equally.

For purposes of standardization, it is essential that definite limiting sizes for both mating members be established which fulfill the foregoing conditions. So far as any one particular standard is concerned, it makes no difference where the dividing line between the two mating parts is made. The essential thing is to have it definitely established in each particular case.

Yet we should not consider any standard as a unit in itself. Another purpose which standardization should serve is to simplify and reduce, as far as possible, stock sizes of material, tools, etc.. The benefits of such a procedure should be self-evident.

Here we must consider two general types of mating surfaces; common or standard, and special. Among the common surfaces are cylindrical, taper and flat surfaces. An example of special surfaces would be splined shafts and fittings.

Where mating surfaces of standard form are involved, for purposes of simplification one limit of one of the mating members should be some basic dimension. Such a basic dimension would be some even number of inches or commonly used fraction of an inch. The tolerance used in such a case is, of necessity, a unilateral one; otherwise the basic dimension chosen would not represent one of the limits but rather something between them.

MAXIMUM SHAFT OR MINIMUM HOLE BASIC

The question next to be answered is, Which part should be made basic? Where the tightest permissible condition has a zero allowance—in other words, when the parts are of the same size—both members will be made basic. In other cases it would be a matter of whether it would be more economical to use a standard size of tool or standard size of material. When a standard size of tool is to be used, the minimum hole should be made basic. When a standard size of material is to be used without machining, the maximum shaft should be made basic.

In a very large number of cases the greatest economy is secured by using tools of standard sizes for the holes. For smaller sizes in particular, these tools are usually solid without any possible adjustment, while the tools for the mating members are usually adjustable. In such cases the logical step is to make the minimum hole of basic size.

On the other hand, shaft members are often used without machining, such as tubing, cold-drawn shafting, etc. In addition, various types of apparatus are often purchased which have connecting members of basic size. Sometimes loose fits and sometimes tight fits are desired on these same connecting members in different installations. In all of the foregoing cases logic would dictate making the maximum shaft of basic size, although special tools would be required to machine the mating members to secure the desired class of fit.

For mating parts of special form, the starting point or nominal sizes can be chosen without regard to anything except the particular form in question. As the tools are special for the one purpose only, no advantage is gained by selecting any exact nominal size. In general, the sizes selected should be governed by the stock sizes of material available from which the shaft member is to be made.

Once definite sizes are selected, however, and if both tight and loose fits are desired with this special form of mating surface, it is a measure of economy to make the sizes of the hole member constant whenever possible, thus requiring but one special tool, and to make the variations needed to obtain the desired fit on the shaft member.

SUMMARY

Summarizing the foregoing we have the following:

For non-mating surfaces the tolerances may be bilateral or unilateral, depending entirely upon the nature of the variations which develop in manufacture. There can be no question here of minimum-hole basic or maximum-shaft basic, as there is but one member to consider.

No mention has been made in the foregoing of the location of holes. Where tolerances are required on such dimensions, usually they should be bilateral, as a variation in either direction is generally equally dangerous. The variation in the distance between shafts carrying mating gears, however, should always be unilateral and plus; otherwise the gears might be too tight to assemble, while a slight increase in the backlash between gears is seldom of such importance. In other words, the tolerance on the distance between holes should be bilateral when variation in either direction is dangerous and should be unilateral when a variation in one direction is less dangerous than a variation in the other direction.

On mating surfaces, with but few exceptions, the tolerances should be unilateral.

The minimum hole should be of basic size in all cases where the simplification and use of standard tools represents the greatest economy.

The maximum shaft should be of basic size in all cases where the use of standard purchased material without further machining, and the provision of special tools for machining the mating part, represent the greatest economy.

One exception to the use of unilateral tolerances on mating surfaces occurs where tapers are involved. In this case either bilateral or unilateral tolerances may prove advisable, depending upon conditions. These should be determined in the same way as the tolerances on distances between holes. When a variation either in or out of the position of the mating taper surfaces is equally dangerous, the tolerances should be bilateral. When a variation in one direction is of less danger than a variation in the other direction, the tolerances should be unilateral and in the less dangerous direction.

METHODS OF EXPRESSING TOLERANCES

Many different methods are employed to express the tolerances on drawings or specifications. The essential consideration in the choice of such a method is to choose the one which offers the least opportunity of being misunderstood. The only purpose of a drawing is to record or convey the necessary information to enable a given piece to be manufactured. The method used to convey this information depends largely upon how it is to be used. Some parts, made in extremely large quantities, are controlled in manufacture entirely by the tools and gages. Sometimes drawings are not used at all in the production departments. In such cases the part drawings can specify limiting dimensions only.

In other cases, where parts are made in smaller quantities, the drawings themselves are the only special facilities employed. In these instances the information should be given in such a manner that the parts can be roughed-out and finished with the least chance of misunderstanding. As an example, we will assume that we have a shaft and hole, nominally 2¼ in. in diameter, and that the minimum clearance is 0.001 in., while the maximum clearance is 0.005 in. Several ways of expressing this information are as follows:

Diameter of hole, 2.250
2.252
Diameter of shaft 2.249
2.247

The foregoing gives the limiting sizes.

Diameter of hole, $2.250 \begin{matrix} +0.002 \\ -0.000 \end{matrix}$
Diameter of shaft $2.249 \begin{matrix} +0.000 \\ -0.002 \end{matrix}$

The foregoing gives one limiting size on each part with a tolerance.

Diameter of hole, $2\frac{1}{4}'' \begin{matrix} +0.002 \\ -0.000 \end{matrix}$
Diameter of shaft $2\frac{1}{4}'' \begin{matrix} -0.001 \\ -0.003 \end{matrix}$

The foregoing method is widely used when parts are made in small quantities and where no gages are used in the

TABLE 1—STANDARDS WITH BILATERAL TOLERANCES

Name of Standard	*S. A. E. Handbook Page*	*Nature of Surface*
Insulated Cable	B33	Atmospheric Fit
Insulated Cable	B36	Atmospheric Fit
Storage Battery Jars	B40	Atmospheric Fit
Plain Washers	C5c	Non-Mating (Thickness)
*Rod-Ends	C9	Mating (Diam. of hole)
Rubber Hose for Gasoline	C51	Non-Mating
Ball Studs	C58b	Taper Diameter
Round High-Strength Steel Wire	C71	Atmospheric Fit
Motorcycle Wheels	F3	(Circumference of Rim)
Motorcycle CC Rims	G9	Fits Rubber Tire
Solid-Tire Base-Bands	G11	Atmospheric Fit
Solid-Tire Felloe-Bands	G12, 13	Atmospheric Fits
Solid-Tire Demountable Rims	G15	Non-Mating
*Spring-Eye Bushings	H5	Mating (Diam. of Eye)
Width of Spring Ends	H6	Non-Mating
Door Locks and Handles	K45	Non-Mating

roughing operations. In many such cases the measuring tools used in the roughing operations are steel scales and calipers only. The use of decimals for the nominal size would be confusing to the operator who is roughing-out material. As he leaves from 1/32 to 1/16 in. for finishing, the few thousandths of an inch specified for the allowance and tolerance is of no importance to him. The operater who finishes the surfaces usually measures them with micrometers, and in measuring the shaft it is as easy, or easier, for him to measure from one to three thousandths under 2¼ in. as it would be to measure from 2.249 to 2.247 in.

Any of the stated methods are entirely suitable for expressing unilateral tolerances. All of them are self-evident.

Another method, which should be discouraged, is sometimes used to express tolerances. This method expresses an intrinsically unilateral tolerance as regards the nominal size as a bilateral tolerance on the mean size. It is as folows:

Diameter of hole, 2.251 ± 0.001

Diameter of hole, $2.151 + 0.002$
$- 0.001$

There is no value in such cases in emphasizing the mean size. It is more liable to confuse than to simplify. Furthermore, the original tolerances specified often are changed as time and experience indicate. In the foregoing example, if the tolerances on both surfaces were increased a thousandth, we would then have the following:

Diameter of hole, $2.251 + 0.002$
$- 0.001$

Diameter of shaft, $2.248 + 0.001$
$- 0.002$

In such cases—and many of them exist—the denoted size is no longer the mean size, and has absolutely no excuse for existence. It is best practice, in all cases where the tolerances are unilateral ones from a standard nominal size, to express them by one of the first three methods shown.

In the case of a bilateral tolerance on a non-fitting surface or atmospheric fit, the tolerance is best expressed as a bilateral one, as follows:

$\frac{3}{4} \pm 1/32$

Sometimes it is desirable to express the dimensions in decimals, as follows:

0.250 ± 0.005

In addition to the methods of expressing tolerances, there is the subject of the proper methods of dimensioning drawings with tolerances so that there can be no misunderstanding. This is such a large subject in itself that no other mention, except calling its existence to attention, will be made now.

SUMMARY

It should be evident from what is said above that there is proper use of both unilateral and bilateral tolerances; of making the minimum hole basic and also of making the maximum shaft basic. Also that varying methods of expressing these tolerances and allowances can be employed to advantage. The attempt has been made to indicate in general their proper application.

S. A. E. Practice in Standards

It is of interest in this connection to review some of the existing S. A. E. Standards, in which tolerances are specified, to see how nearly they already agree with the practice proposed in this discussion. Possibly some of the standards have been overlooked in making this summary, but practically all of them have been included.

There are 46 standards wherein unilateral tolerances only are used. There are 17 standards in which bilateral tolerances only are used.

There are 27 standards in the case of which both unilateral and bilateral tolerances are used, sometimes on different surfaces of the same standard part and sometimes on the same surfaces of different sizes in the series.

The best practice is to use unilateral tolerances only on mating surfaces, while either unilateral or bilateral tolerances can be used on non-mating surfaces. We will therefore review in detail only the bilateral tolerances.

TABLE 2—STANDARDS WITH UNILATERAL AND BILATERAL TOLERANCES

Name of Standard	*S. A. E. Handbook Page*	*Nature of Surface Where Bilateral Tolerance Is Used*
Lamp Glasses	B6	Non-Mating (The Three smaller sizes use unilateral tolerances only)
*Magneto Mountings	B14a	Mating (Shafts and Coupling Bore)
Generator Mountings (Flange Type)	B15a	Non-Mating
Annular Ball-Bearings (Open)	C32	Non-Mating (Width. Note, on several other types the tolerance is unilateral on this surface also)
Annular Ball-Bearings	C33a	Non-Mating (Width)
Annular Ball-Bearings	B33b	Non-Mating (Width)
Annular Ball-Bearings	C33c	Non-Mating (Width)
Thrust Ball-Bearings	C40	Non-Mating (Thickness)
Thrust Ball-Bearings	C41	Non-Mating (Thickness)
Thrust Ball-Bearings	C42	Non-Mating (Thickness)
*Roller Bearings	C43	Mating (Bore) Non-Mating (Width)
*Roller Bearings	C44	Mating (Bore) Non-Mating (Width)
Fuel and Lubrication Pipe Fittings	C47	Non-Mating
Ball Studs	C58c	Taper Diameter
Plain Hex-Head Bolts and Nuts	C60, 61	Non-Mating
Ball Hex-Head Bolts and Nuts	C62, 63	Non-Mating
*Tachometer Drive	C75	Mating (Square Broached Hole Non-Mating
Tachometer Drive	C76	Non-Mating or Clearance
Tachometer Drive	C77	Clearance
Turnbuckles	C79	Atmospheric Fit
*Turnbuckles	C81	Mating (Thickness of Eye) Atmospheric Fits
Roller-Chain Sprockets	E4	Clearance
Three-Joint Propeller-Shafts	E6b	Diam. of Taper
Clutch Facings	E19	Location of Holes Non-Mating (Thickness)
Motorcycle Spokes	F4	Atmospheric Fits
Pneumatic-Tire Rim-Sections	G3	Non-Mating (Fits Tire)
Felloe-Bands	G7	Non-Mating

Of the 17 standards in Table 1 only two have bilateral tolerances on mating surfaces. These items are starred in the tabulation; namely, Rod-Ends and Spring-Eye Bushings. It would seem advisable to have them corrected at the earliest opportunity.

The standards in Table 2 use bilateral tolerances and also unilateral tolerances. The tabulation indicates the nature of the surface on which the bilateral tolerances only are used. Of the foregoing 27 standards, only 5 have bilateral tolerances on mating surfaces. These items are starred in the foregoing list. Thus, out of 44 standards where bilateral tolerances are used, only 7 need be changed in order to bring them all in agreement with the practice recommended.

There are 21 standards where the minimum hole is basic.

There are 12 standards where the maximum shaft is basic.

There are 20 standards where both mating parts start at basic size.

There are several standards in which neither member is of basic size. Most of these are of special form. In a few cases, where both tight and loose fits are specified, neither member is of constant size, so that tools and gages are special, not only for each size but also for each class of fit. This is an extremely uneconomical practice and should be corrected.

This discussion and brief summary of S. A. E. Standards is submitted for comment and criticism. It seems very desirable to bring this matter before the Society at this time in order to work out some uniform practice to which all of the Society Standards may adhere.

Whatever practice is adopted, it should be definite enough to avoid confusion, and also be flexible enough to meet the many varying conditions with which the industry as a whole must contend.

SIMPLE BRAKE TEST NEEDED

Activity of Police in Inspecting Brakes Warrants Adoption of Standard Brake Test

Fourteen recommendations to decrease automobile accidents were made in the first report of a special committee appointed by the Grand Jurors' Association of New York City to study the causes of the steady increase in deaths due to motor cars. The suggestions were the outcome of a conference with officials concerned with the automobile trade, traffic and the enforcement of the rules, regulations and ordinances covering automobile operation. The fifth recommendation proposed a strict examination for defective brakes.

The test used by the New York police safety-squad engaged in testing automobile brakes is as follows:

> The officer takes a seat by the driver and directs that he speed up the car to between 15 and 18 m.p.h. Then at his direction the driver applies the foot-brake. If the car is stopped within 30 ft. the foot-brake is considered satisfactory and the driver is instructed to repeat the performance, but using his emergency brake. If either brake is found defective the driver is ordered to return within three or four days for a check-up. If both brakes are found defective the driver receives a summons to appear in court.

The tests are generally conducted on the city streets, irrespective of the condition of the pavement with the exception that wet streets are avoided.

The Police Department does not look with favor upon the testing of brakes by machines developed for that purpose. They feel the only satisfactory test is a service test made under normal conditions. The opinion of certain New York police officials in reference to present-day design are of particular interest. They feel that car manufacturers have paid no attention to brake design for the last 10 years. They consider brakes to be the most important unit in the car, owing to their importance as a means of avoiding fatalities in traffic. They feel that, with the increasing practice of cleaning streets by flushing, together with the ever-present danger of wet streets due to rain, brakes should be designed which will meet such conditions.

This subject is receiving the careful consideration of the Subdivision on Brake Tests, of which Clarence Carson of Dodge Brothers is chairman. Comments received from committee members in reference to the New York practice of testing brakes indicate that the requirements are considered fair, but that a definite standard should be promulgated which might be followed generally throughout the Country. The following comments received are of special interest.

Stopping Distances

In general we believe that any brake tests should consist of an actual driving test, preferably on a dry road surface such as concrete where the friction between the wheels and the road is fairly constant. Cars should be tested with full passenger-loads if possible. We expect our touring cars to stop under the above conditions in the following distances:

Car Speed, M.P.H.	Stopping Distance, Ft.
10	6
15	12
20	28
25	38
30	65
35	80
40	120

We would consider the above distances as shown to be rather more exacting than could be expected from the ordinary rear-wheel brake equipment, but are giving them so they may be considered as the best that can be done without using front-wheel brakes and one that is commercially practical with a well-designed transmission type of service-brake.

Test Conditions

The stipulation of bringing a car to rest within 50 ft. at 20 m.p.h. seems very reasonable indeed. We know that properly designed rear brakes can always be relied on to do better than this on any reasonable surface.

The present allowance for a variation in speed between 15 and 18 m.p.h., implying a variation of 20 per cent in speed and 44 per cent in stopping distance, is entirely too inaccurate. Accurate means should be available for checking the speed of the car.

The testing track should be under cover, as surface conditions, especially on tarred roads, vary considerably with the temperature as well as with moisture. The tire pressure should be stipulated and should be that recommended for use by the makers of the vehicle, rather than the high pressures called for by some of the tire manufacturers.

A standard loading should be provided, which might reasonably consist of the full complement of passengers for two and three-passenger machines, and four persons for larger passenger cars (150 lb. to be allowed for each passenger).

Nature of Test

In the test of brakes on cars we have not yet found any satisfactory mechanical means for doing the work and believe that a road test is the best method at the present time. This road test can be carried out by any person of average intelligence and requires no special technical knowledge on the part of those conducting the test.

Under average dry road conditions the stopping distances when running at the rate of 20 m.p.h. should be 37 ft. To allow for possible nervousness on the part of the car operators when undergoing this test, we have usually allowed 10 per cent over this distance, calling 40 ft. satisfactory for stopping the car when going at 20 m.p.h. In conducting these tests it is usually well to have some mechanic as an observer to note the adjustment of the bands on the wheels and

the distance the foot-pedal goes down toward the floorboard when applying the brakes.

We know that this test works well and it can be carried on very expeditiously by any police force.

Brake Surface Area

We are heartily in favor of brake tests, not only on account of the number of brakes found defective in tests we have observed, but also because of the number of drivers we have found who do not know how properly to bring their cars to a stop.

It seems to me that more attention is being paid to the brakes on the wheels in particular than to the entire braking system in general. The ratio of square inches of braking surface to 100 lb. of car weight varies through a great range and it seems to me that this is a matter which could be standardized. There is also a great variation of brake-drum diameters. I believe this matter is one which should receive some consideration and the brake-drum diameter, together with its width, should bear a certain relation to the car weight.

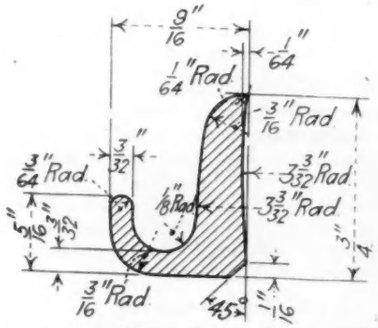
I feel that some study could be profitably given to the linkage system and the pressure required on the foot-pedal to produce the desired deceleration. The number of women drivers is increasing every day and a woman cannot exert the pressure on a foot-pedal that a strong man can. Consequently, I believe that the brake system should be so designed that the brakes can be effectively applied by the average woman driver.

REVISED MOLDING STANDARDS PROPOSED

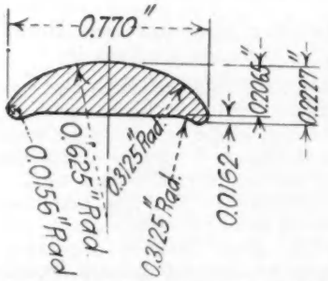
Original Recommendation Covering Aluminum Molding Changed as a Result of Comments

The progress report which was published in full in the May issue of The Journal has been revised as a result of criticism received from body builders and passenger-car manufacturers building bodies. It has been found that the demand for ell and tee molding is so limited that it is inadvisable to attempt to standardize any particular styles. On the other hand, the use of belt molding being so extensive, it was decided that more than one type should be included in the standard. The recommendation has therefore been revised to include one type of drip molding and three types of belt molding, there being three sizes for one of these types. The complete report thus covers six types and sizes of molding. The report is given in full in the accompanying illustrations and table.

It should be appreciated that the moldings proposed are standard at the present time, having been selected from those sizes most widely used in present practice. As the molding manufacturers are called upon to furnish an almost unlimited number of types and sizes of molding, practically no two orders of which specify the same cross-section, the advantage to the industry if the proposed types of molding are adopted as standard practice will be appreciated.



Estimated Weight per Linear Foot: 0.180 Lb.
Fig. 1—Drip Type of Molding



Estimated Weight per Linear Foot: 0.143 Lb.
Fig. 2—Type A Belt Molding

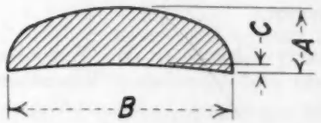
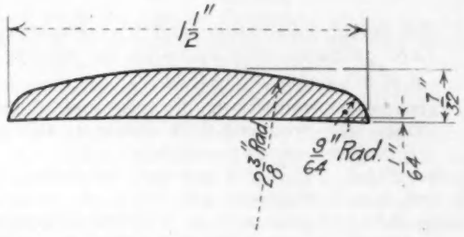


Fig. 3—Type B Belt Molding

No.	A	B	C	Estimated Weight Per Foot, Lb.
B1	¼	⅞	0.025	0.186
B2	¼	1	0.025	0.228
B3	¼	1¼	0.025	0.286



Estimated Weight per Linear Foot: 0.331 Lb.
Fig. 4—Type C Belt Molding

PROPER SPARK-PLUG SPACE IMPORTANT

Minimum Clearance Dimension to Be Determined as a Guide for Future Engine Design

The Spark-Plug Subdivision of the Electrical Equipment Division has indicated that it would be desirable to establish a standard minimum dimension for the vertical distance from the spark-plug gasket-seat to the nearest interfering object. This dimension is considered important as there is a tendency on the part of some manufacturers to make this clearance such that certain types of spark-plug which the public demands cannot be used because of lack of proper clearance.

In order to determine whether it is feasible to standardize such a dimension, information has been obtained on present practice and turned over to the Spark-Plug Subdivision. It is expected that a definite recommendation based on this information will be submitted at an early date.

IRON AND STEEL DIVISION ACTIVE

Physical-Property Charts to Be Extended to Include Chromium and Chromium-Vanadium Steels

The charts of physical properties given in the S. A. E. Handbook are considered as indications of what may be expected of the average product of a given range in analysis when treated in 1½-in. sections, from which tests are taken from standard 0.505 x 2 in. specimens. Quoting from page D18 of the S. A. E. Handbook:

> For the sake of simplicity it has seemed advisable to use only average minimum figures for tensile-strength, yield-point, reduction of area and elongation, based on the following considerations, heat-treatment being constant:
>
> (1) The lowest yield-points and tensile-strengths are produced with steels at the bottom of a given range in carbon
>
> (2) The lowest reductions in area and elongations are produced with steels at the top of a given range in carbon
>
> Thus, for Steel 1035, the average minimum yield-points and tensile-strengths are given as of a steel containing 0.30 per cent carbon; average minimum reductions of area and elongations as of a steel containing 0.40 per cent carbon

The results given in the present charts were estimated from accumulated data, in most cases on sizes under 1½ in. round.

The Subcommittee on Chromium and Chromium-Vanadium Steels feels that data of this kind for these types of steels will not give entirely satisfactory information on material covering a 10-point carbon-range in any of the alloy steels and will be of no practical value for the chromium steels covered by the committee.

In the S. A. E. charts, while the lowest tensile-strength and elastic-limit apply on steels at the bottom of a given carbon-range, and the elongation and reduction on steel at the upper limit in the carbon-range, no consideration whatsoever is given to the alloying elements. The minimum average values, to be of practical use, should take this point into consideration. Whether the alloying elements are on the high or low side of the range has tremendous influence on the physical properties developed. For instance, on S. A. E. Steel 3130 a wide variation in results would be obtained on a 1½-in. round from two analyses as follows: Carbon, 0.25; Chromium, 0.45; Nickel, 1.00 (lower extremity of the range in alloys) and Carbon, 0.25; Chromium, 0.75; Nickel, 1.50 (upper extremity of the range in alloys). The first-mentioned analysis will not by any means meet the results in tensile strength as indicated in the graphs of average minimums on a 1½-in. round with an 800-deg. fahr. draw. Likewise, on S. A. E. Steel 3120 and various other types, results are much too high to be representative of average minimums with the limits on the lower extremity of the range.

For chromium and chromium-vanadium steels, the difficulties in giving graphs of practical value in accordance with the present method employed are greatly increased. The reason for this lies in the very nature of the alloying element or elements, where the increase in physical properties resultant from these elements is due directly to the formation of complex carbides, of which these alloying elements are components.

The recommendation of the Subcommittee on Chromium and Chromium-Vanadium Steels is as follows:

> That graphs be given showing actual results obtained on a definite size (1⅛-in. round has been decided upon as representative of requirements) of definite chemical properties as indicated (within range under consideration) and with graphs developed from actual tests on such material

The recommendation of the Subcommittee has been submitted to the members of the Iron and Steel Division and metallurgists connected with the different automobile companies for comment in order that the Committee may be confident that when the data are compiled they will be of the most value to the users thereof.

BRAKE-DRUM STANDARD SUBMITTED

Future Requirements Met by Eight Drum-Diameters, But Only Maximum Width Specified

The Subdivision on Brake-Drums has submitted to the Axle and Wheels Division a final report covering the standardization of brake-drums. The recommendation, which is given in the accompanying table, will be acted upon at the next meeting of the division and, if approved, submitted to the Standards Committee for action at the Annual Meeting.

BRAKE-DRUM DIMENSIONS PROPOSED AS STANDARD

No.	Outside Diameter	Inside Diameter	Stock Thickness	Maximum Height of Wall	Maximum Eccentricity
0 (Ford)	$8\frac{15}{64}$	8.005– 7.995	0.1250	$1\frac{5}{16}$	0.020
1	10	9.790– 9.770	0.1250	$1\frac{5}{8}$	0.020
2	11	10.710–10.690	0.1650	$2\frac{1}{2}$	0.020
3	12	11.666–11.646	0.1875	$2\frac{1}{2}$	0.020
4	13	12.666–12.646	0.1875	$2\frac{1}{2}$	0.020
5	14	13.635–13.615	0.2030	$2\frac{3}{4}$	0.020
6	15	14.635–14.615	0.2030	3	0.020
7	16	15.605–15.585	0.2180	$3\frac{1}{2}$	0.030
8	17	16.541–16.521	0.2500	$3\frac{1}{2}$	0.030

Although the Subdivision voted favorably on the report, some comments were submitted which will be considered by the division when final action is taken, the principal criticism being that maximum brake-drum widths, rather than definite widths, have been specified. It is apparently felt that the standard would be of more value if a definite width or series of widths were specified for each brake-drum diameter. Comments in reference to this point would be appreciated by the division members.

LEATHER GRADE NAMES CONFUSING

Upholstery Leather Specifications Being Prepared to Obtain Uniformity

At the present time trade practice as regards the nomenclature of the various splits of hides varies to such an extent that it is almost impossible to determine what split is designated by a grade name. There is also a total lack of uniformity in the leather specifications used by the various companies. Appreciating that the formulation of definite specifications would be of extreme value to the body industry, the Passenger-Car Body Division appointed a Subdivision some time ago to recommend definite specifications for upholstery leather. K. L. Herrmann, of the Studebaker Corporation, was appointed chairman, the other members being F. E. Watts, Hupp Motor Car Corporation; W. H. Graves, Packard Motor Car Co.; W. N. Davis, Cadillac Motor Car Co.; and R. K. Mitchell, Maxwell Motor Co., Inc.

Subsequent to the appointment of this Subdivision, R. C.

Bowker, of the Bureau of Standards, was appointed to the Subdivision in view of the Bureau's work in developing specifications for upholstery leather for Government use.

Information was obtained from the various leather users as to the specifications they were purchasing to, but it was found that considerable study would have to be given to the subject. It was, for instance, found that many companies would not accept hides patched with leather of the same grade or thickness and skived so as to be unnoticeable from the finished side. Study of this problem indicated that the use of certain glues would make this practice satisfactory to leather users. As the result of close cooperation between the Subdivision and the Bureau of Standards, information has been obtained by the Bureau which is of considerable interest in the connection. This information has been issued as Letter Circular LC 97 of the Bureau of Standards and is given hereinafter in full with the exception of illustrations showing the detailed results for full-grain and machine-buffed splits.

Notes on Upholstery Leather

In connection with the development of specifications for the different grades of upholstery leather for Government use and as a result of a request for cooperation on the same subject from the Passenger-Car Body Division of the Society of Automotive Engineers, certain investigations have been made relative to standards which will serve as a measure of the quality of these leathers. The procedure followed

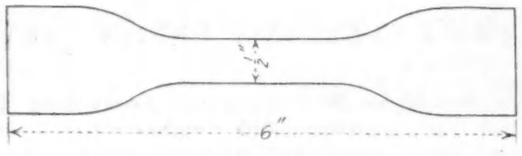


Fig. 6—Type of Tensile Test-Specimen Used

and the results obtained in this investigation are presented as a basis for the selection of satisfactory specification requirements, and also to stimulate further interest and work on a subject regarding which few data are available.

Definitions of Grades.—Since some confusion has existed regarding the meaning of the commercial classifications of grades, the following is suggested as a possible standard which covers the three grades mostly used by automobile manufacturers:

Grade	*Definition*
Full-Grain (Snuffed)	The first split from a hide containing the full grain of the leather, the surface of which is shaved either by hand or machine, removing shavings of no appreciable size or thickness.
Machine-Buffed	The first split from a hide after a thin layer of the grain thickness (called a buffing and approximately 1/64 in. thick) has been removed.
No. 1 Split	The second split or thickness of a hide secured after either a full-grain or machine-buffed split has been removed. (The No. 1 Splits under both a full-grain or a hand-buffed split appear to be near enough equal in quality to have the same classification although in the latter the cut is begun approximately 1/64 in. deeper.)

Selections.—One of the chief causes of trouble in the use of leather for upholstering is the number of defects in the hides. There has been little attempt to make a standard selection to which all tanners will adhere. Most of these defects are caused by circumstances entirely outside of the tanner's control and it would naturally be more to his interest to sell his leather on the basis of "tannery run." In the case of the consumer, where production is important and hides are cut on a piece-work basis, it appears essential that the selection of leather being used at any one time be as uniform as possible. The following specifications as to selections have been suggested:

A—Selection shall consist of leather free from brands, patches, knife-cuts, knife-scores and open grub-holes, excepting that 10 per cent of the hides may contain not over 4 open grub-holes

B—Selection shall consist of leather free from brands, knife-cuts and knife-scores. Hides may contain not over 4 patches 1½ in. in the longest dimension and not over 9 open grub-holes

No attempt has been made to cover the various other selections which may contain various types of brands and other imperfections. It is believed that a large portion of the leather required can be furnished under the above and that, where a consumer finds he can use lots with certain types of brands or more of the defects than allowed in the above, negotiations can be made with the contractor regarding a definition of the selection he will purchase.

Shapes and Sizes of Hides.—An important factor is the shape of the hides since they will generally cut to better advantage the nearer they are to being square in shape. The shape is governed chiefly by the type of animal from which the hide comes and to a less degree on the manner of skinning. Where hides of nearly uniform width are required the width across the brisket must be specified. With reference to area the idea at one time seemed to be to get as spready hides as possible. However, the Leather Subdivision of the Passenger-Car Body Division of the Society of Automotive Engineers has already demonstrated that smaller hides can often be cut to advantage and in fact to such an extent that the tanners will have no cause for the complaint that hides of large area are always demanded. Specifications for maximum, minimum and average area in square feet must be specified by the purchaser according to the use of the material.

Thickness.—The common practice among tanners is to gage the thickness by ounces. Since it is stated that an ounce is equal to 1/64 in., the following is suggested as a thickness specification that can be readily understood by all:

> The leather shall be 3/64 in. in thickness plus or minus one half of 1/64. The thickness shall be measured with a Woburn gage or equivalent

Cracking of Coating.—The cracking of the coating is an important factor in the use of upholstery leather and it is suggested that the coating should show no evidence of cracking when the leather is doubled on itself by hand, coating side out, at 32 deg. fahr.

Fiber Appearance.—With reference to fiber appearance, a vertical section of full-grain leather shall show approximately 50 per cent of the thickness to be made up of the fine

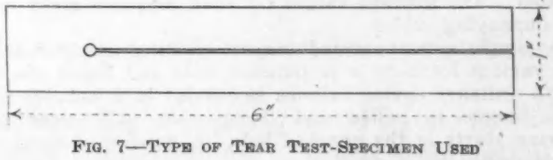


Fig. 7—Type of Tear Test-Specimen Used

fibers and a vertical section of machine-buffed leather shall show approximately 25 per cent of the grain fibers.

Physical Test Data.—In order to determine the magnitude of the physical properties of upholstery leather 10 hides were submitted by automobile manufacturers for test. The hides submitted are listed below according to grade and code number.

Grade	*Code Number*
Full-Grain	A, B, C, H and I
Machine-Buffed	D, F and J
No. 1 Split	E and G

One-half of each hide was divided into blocks approximately 6 by 12 in. From each block three samples were cut

with a die for tensile tests. The percentage of stretch at failure in a gage length of 2 in. was noted on the same samples. The average of the three values was used as representing the value for the block. Three tear test-specimens were also cut from each block. One test-specimen from each block was used for the grab tear-test and the stretch at failure was also noted for this sample.

The grab-test sample was 3 by 6 in. and was placed length-

AVERAGE PHYSICAL PROPERTIES DETERMINED FROM TEST-SPECIMENS CUT LENGTHWISE OF THE HIDES

Hide	Breaking Strength, Lb.	Tensile Strength, Lb. per Sq. In.	Percentage Elongation at Breaking Point Gage Length 2 In.	Tensile Strength for Grab Test, Lb.	Percentage Elongation at Failure in Grab Test	Tearing Strength, Lb.	Average Thickness, In.
A	48	2350	34.0	209	42	7.3	0.041
B	52	2105	39.7	195	41	9.4	0.050
C	59	2990	38.0	223	41	8.0	0.039
H	50	2490	38.0	168	41	7.8	0.041
I	39	2210	42.1	156	36	7.2	0.036
D	58	2635	39.6	238	46	14.8	0.044
F	60	2950	47.1	229	51	15.1	0.041
J	45	2140	34.7	194	36	12.1	0.042
E	51	2470	36.1	233	39	16.6	0.041
G	63	2645	37.4	309	39	21.2	0.047

AVERAGE PHYSICAL PROPERTIES DETERMINED FROM TEST-SPECIMENS CUT CROSSWISE OF THE HIDES

Hide	Breaking Strength, Lb.	Tensile Strength, Lb. per Sq. In.	Percentage Elongation at Breaking Point Gage Length 2 In.	Tensile Strength for Grab Test, Lb.	Percentage Elongation at Failure in Grab Test	Tearing Strength, Lb.	Average Thickness, In.
A	45	2170	43.0	163	49	7.0	0.042
B	51	2010	39.0	201	41	8.3	0.051
C	53	2665	38.0	212	40	7.2	0.040
H	53	2460	37.0	186	36	6.7	0.043
I	40	2260	39.2	160	38	7.4	0.036
D	56	2615	38.0	225	42	14.6	0.043
F	55	2715	53.0	226	53	16.5	0.040
J	41	1900	34.7	190	37	13.3	0.042
E	41	1985	39.0	204	41	19.1	0.041
G	64	2740	36.0	292	39	21.5	0.046

wise in the testing machine, the clamps being 3 in. apart. The upper jaw of each clamp was 1 in. in width and the lower jaw 3 in. The test-specimens for tensile strength and tear are shown in the accompanying illustrations.

Beginning with block No. 1 at the tail, the test-specimens were cut in a lengthwise direction. Samples from alternate blocks were cut in the same direction, while those from the remaining blocks were cut in the crosswise direction. The data obtained give results of the variation in physical properties for different locations on the hide in both the lengthwise and the crosswise directions and for three grades of leather. The average values for each hide are given in the accompanying tables.

A simple test suggested for determining the strength of the various leathers is to punch a hole and insert the hook of an ordinary spring-balance suspended in a suitable place. The sample is pulled and the pounds pull noted when tearing starts in the punched hole. A number of tests were made using this method.

Summary.—The results show that the greatest variation between the different grades is not in tensile-strength, but in a test which involves tearing, the grab test or the test with the spring-balance. For laboratory examination it is

AVERAGE PHYSICAL PROPERTIES DETERMINED FROM TEST-SPECIMENS CUT LENGTHWISE AND CROSSWISE OF THE HIDES

Hide	Breaking Strength, Lb.	Tensile Strength, Lb. per Sq. In.	Percentage Elongation at Breaking Point Gage Length 2 In.	Tensile Strength for Grab Test, Lb.	Percentage Elongation at Failure in Grab Test	Tearing Strength, Lb.	Average Thickness, In.
A	47	2260	38.5	186	46	7.1	.....
B	51	2060	39.3	198	41	8.8	.....
C	56	2830	38.0	217	41	7.6	.....
H	52	2475	37.5	177	39	7.2	.....
I	40	2235	40.5	158	36	7.6	.....
D	57	2625	38.8	231	44	14.7	.....
F	57	2835	50.0	227	52	15.8	.....
J	43	2020	34.7	192	36	12.7	.....
E	46	2225	37.5	219	40	17.8	.....
G	63	2690	36.7	300	39	21.4	.....

considered that the grab test is most satisfactory with the following limitations on the results for the different grades regardless of the location on the hide.

Grade	*Minimum Pounds by Grab Test*
Full-Grain	100
Machine-Buffed	150
No. 1 Split	200

These requirements are conservative, but sufficient to show that a sample of leather has the strength necessary for the service intended and also to eliminate tender hides.

SMALL CLEARANCE UNNECESSARY

Larger Bolt-Holes Will Facilitate Assembling Transmission and Engines

As a result of the following comments received by the Society from an engine manufacturer, the standardization of the size of the transmission mounting bolt-holes for the flywheel-housing type of construction was referred to the Transmission Division:

> We have occasionally received complaints that it is a difficult matter to line-up the transmission mounting holes with the tapped holes of the flywheel housing so that the capscrews can be inserted properly. Inasmuch as the holes in the flywheel-housing flange are tapped and will vary slightly, the holes in the transmission flange are in most cases only 1/64 in. larger than the capscrews used
>
> However, these capscrews have no aligning purpose, as a transmission flange has a pilot which fits in the flywheel housing. We can therefore see no adequate reason why these holes should not be made larger. The present practice of some transmission manufacturers is to use a 1/64-in. clearance, others use a 1/32-in. clearance and some, the writer understands, use a clearance as large as 1/16 in.
>
> We can see no objection to a 1/16-in. clearance, as it will facilitate attaching the transmission to the engine and would not affect the holding quality of the joint

In order that the Transmission Division may take proper action in respect to this suggestion, information has been obtained by the Society as to what clearances are used at the present time. It is expected that a report will be submitted at the Annual Standards Committee Meeting covering this subject.

SOCIETY MEETINGS

PRODUCTION MEETING OCTOBER FEATURE

Two-Day National Meeting in Cleveland Expected to Attract Large Attendance

Four meetings crammed with valuable production information, intensely interesting factory visits and an informal Production Dinner constitute the features of the Second National Production Meeting of the Society which will be held in the Winton Hotel, Cleveland, Oct. 25-26. Over 600 production men received the 1922 Production Meeting in Detroit enthusiastically because it gave them new ideas to carry home. The program of the Cleveland Meeting has been planned along exactly these lines; practical usable information will be found in all of the papers. There are ten papers assured for the meeting and these are described briefly in the lists of speakers in another column. A brief study of this program should convince production executives that it will be profitable for them to drop all factory ties and become a part of the Production Meeting.

From the opening of the first session on Thursday morning to the final word at the Production Dinner on Friday evening, all thought will be concentrated on matters of interest to the factory executive. Topics listed on the program are sufficiently diversified to hold interest for nearly all classes of shop executives. Time will be provided for questions and answers following each paper. Some time will also be available for general discussion of the more important topics.

A special effort will be made to start all of the meetings promptly. They will be held in the Rainbow Room of the Winton Hotel, which is especially suited to the staging of meetings of this type. The banquet on Friday evening will be held in the ballroom of the Winton Hotel. Everyone interested in the advancement of automotive production methods is welcome at the meetings; it is not necessary to be an S. A. E. member to secure the privilege of attendance at these particular sessions.

Although all details of the program arrangement have not been set definitely at this time, the following outline will be adhered to:

THURSDAY, OCT. 25

9:30 A. M.—Machine Tool Session
1:00 P. M.—Lunch
2:15 P. M.—Factory Managers' Session
8:00 P. M.—Scientific Management Session

FRIDAY, OCT. 26

9:30 A. M.—Inspection Session
1:00 P. M.—Lunch
2:00 P. M.—Factory Visits
6:30 P. M.—The Production Dinner

HOTEL RESERVATIONS

Cleveland hotel facilities will be taxed during the period of the Production Meeting due to the presence of other large conventions besides the S. A. E. Meeting. It is extremely important that you write to the Hotel Winton and make reservations in advance if you wish to be assured of comfortable quarters. Rates are as follows:

Room with private bath, single, $3.00, $3.50, $4.00, $4.50, $5.00
Room with private bath, double, $5.00, $5.50, $6.00, $7.00, $8.00
Room with private bath, twin beds, $6.00, $7.00. $8.00, $10.00

THE PRODUCTION DINNER

Nothing promotes interchange of experience and the development of new ideas more than social contact between individuals engaged in the same field of endeavor. Broadening of acquaintanceship between men in the production end of the automobile business can best be accomplished by functions such as the Production Dinner. That is the entire purpose of the Production Dinner during the Cleveland Meeting. It will be informal in every respect. There will be entertainment and music while the food is being served.

THE PRODUCTION DINNER

APPLICATION FOR TICKETS

1. All Seats are reserved. Preference as to location will be accorded applications in the order of their receipt.
2. Tables seat eight. If you wish to make up your own party, send in one application for the entire group of tickets. All tickets requested on a given application will be for the same table.
3. The dinner is to be informal and the wearing of evening clothes discouraged.
4. No tickets can be cancelled or refund granted after Tuesday, Oct. 23.
5. Orders for tickets mailed after Monday, Oct. 22, should be addressed to the Society of Automotive Engineers, Winton Hotel, Cleveland, O. Tickets will be on sale at the hotel until 6 P. M. Oct. 26.

(See Over)

PRODUCTION MEETING SPEAKERS

H. P. HARRISON
H. H. Franklin Mfg. Co.

Application of conveyor equipment to automobile production. Examples cited of successful installations and others which proved inefficient. Valuable suggestions for use of those responsible for material-handling work.

E. BOUTON
Chandler Motor Car Co.

The regulation of straight piece work and group plan of wage payment. No wage system can be applied to all plants; that which gives the most satisfactory results should be adopted. Description of system in successful operation at the Chandler plant.

OSCAR A. KNIGHT
Norton Company

Developments in production grinding in the automotive industry. Examples of the latest adaptations of the grinding process. Illustrated with instructive photographs.

RALPH E. FLANDERS
Jones & Lamson Machine Co.

The advantages and limitations of standard machine tools for continuous production. The machine tool builders' views on the special vs. general-purpose machine tool. When the application of general-purpose machines becomes uneconomical.

ALBERT F. SHORE
Shore Instrument & Mfg. Co.

Valuable instruction on the use of the scleroscope in studying material hardness. Variations that occur in the use of the scleroscope in the inspection department and how to correct them.

E. P. BLANCHARD
Bullard Machine Tool Co.

Description of two special purpose machine tools of high productivity. Fitting them into a production plan. Accompanied by informative motion pictures.

W. F. JAMESON
Cleveland Automobile Co.

The human element in production. Practical philosophy on the handling of men. Development of the worker's interest in his daily task. Stimulating the worker's powers of concentration to overcome "human element errors."

PERCY S. BROWN
Corona Typewriter Co.

Comprehensive paper covering every detail of production control as utilized in the Corona plant. Explains organization system, theory of their production problem and shows how this is met. Description of ledgers, forms, schedule boards, and string graphic control boards.

A. J. OTT and C. L. OTT
American Grinder Co.

Description of a process of grinding gear teeth of the involute form to contours of high accuracy. Methods of inspecting gear teeth for proper form, tooth thickness and spacing.

A. H. FRAUENTHAL
Chandler Motor Car Co.

Factors governing the measurement of "out-of-roundness." Various adverse conditions are found in measuring size variations of cylindrical objects with modern inspection apparatus. Points that have been determined by a close study of this operation will be presented.

Dinner talks will be short and to the point; at the same time, they will be highly inspirational. The dinner program will be arranged so that plenty of time will be available for friendly chats after the final drop of the toastmaster's gavel.

All seats at the dinner will be reserved. Tickets will sell at $3.50 each and preference as to location will be accorded applications in the order of their receipt. Members who are desirous of reserving tables or individual seats should forward the blank at the bottom of this page with their check to the New York office at once.

Final Announcement

Final announcement of all of the details of the Production Meeting will be included in an issue of the Meetings Bulletin which will reach the entire membership about Oct. 15. This Bulletin will be sent to many production executives who are not Society members. Mail us the names of men in your shop.

THE PRODUCTION DINNER

APPLICATION FOR TICKETS

Society of Automotive Engineers, Inc.
29 West 39th St., New York City.

Order of Receipt.....

Gentlemen:—

Please mail me .... tickets for the S. A. E. Production Dinner at the Winton Hotel, Cleveland, Oct. 26. I am enclosing remittance of $..... to cover the cost of these tickets at $3.50 each.

I understand that all dinner seats are reserved and that preference as to location will be accorded applications in the order of their receipt.

I also understand that dinner tickets are not subject to cancellation or refund after Tuesday, Oct. 23.

....Mail tickets to address given herewith.

....Hold tickets for me at the door.

Signed .....

Address .....

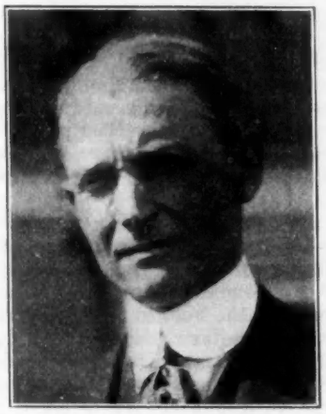
.....

(over)

VINCENT DISCUSSES EIGHT-IN-LINE

Four-Wheel Brakes Also Treated in Talk Before Cleveland Section

Eight-in-line engines, four-wheel brakes and easy steering were the subjects of a comprehensive engineering and mathematical analysis presented by Past-President J. G. Vincent and W. R. Griswold before the Cleveland Section Sept. 17. Attendance records for Cleveland Section meetings were broken on this occasion, there being over 300 present during the technical session and over 150 seated for dinner. The Vincent-Griswold paper, which presents the engineering reasoning underlying the development of the Packard single-eight car, will be found on p. 257 of this issue of THE JOURNAL. The mathematical appendices to the paper will appear in the November issue. Important points raised in the discussion are recorded in the following paragraphs.



J. G. Vincent

Numerous written questions were presented to Mr. Vincent on cards that had been passed through the audience. Answers to these questions occupied a greater part of the discussion period. It developed that a majority of the questions related to four-wheel brakes, the members apparently displaying a greater interest in this feature than in the eight-in-line engine. Asked whether he had considered using only the front brakes for service purposes, Mr. Vincent replied that all the braking effort that could reasonably be gotten was wanted in the service brakes since these are the real emergency brakes, the so-called emergency brake being nothing but a standing brake. In a car that has the ability to accelerate quite rapidly and weighs considerable, maximum brake effectiveness should be provided and obviously it cannot be gotten on two wheels. Also, by distributing the braking effort over four wheels wear is greatly reduced and frequency of adjustment is decreased. In the Packard layout the rear-wheel brakes are also used for the standing brake, but they are operated through an independent mechanism.

BRAKE PRESSURE DIVIDED UNEQUALLY

Pressure is not applied equally to the Packard front and rear brakes, but is divided in the ratio of 45 per cent on the front brakes and 55 per cent on the rear. However, both brakes are applied simultaneously. A maximum rate of deceleration of 20 ft. per sec. can be attained and this effects a stop in 200 ft. from 60 m.p.h. Mr. Vincent remarked that four-wheel brakes take hold more smoothly than the conventional rear-wheel system and rear-axle chatter is avoided. He did not favor propeller-shaft brakes for general service work. He said that very satisfactory four-wheel brakes can be built for the same money as a practical propeller-shaft brake with sufficient strength provided in the rear axle to carry the braking loads.

Mr. Vincent favored internal expanding brakes over those of the external contracting type for several reasons. In the first place, they are housed-in and protected from dirt, and this prevents excessive wear. He believed that external contracting brakes are likely to freeze-up under certain conditions. External brakes are more difficult to adjust, in his opinion, since there are more parts to keep in adjustment, such as the sliding block, the throw, and the top and bottom clearance. He did not believe that it was impossible to design internal brakes so that suitable leverages could be obtained to guarantee efficient brake operation.

FREE-WHEEL THEORY DISPUTED

Some front-wheel braking systems are designed so that the outside front wheel brake releases automatically on a turn, allowing this one wheel to run freely. Mr. Vincent stated that he could see no particular advantage in this arrangement. Geometrical relation of front-wheel brake parts should be such that spring deflection under the braking loads will tend to reduce the braking pressure rather than increase it. This will avoid a violent front-axle chatter that might otherwise be set-up.

The steering-knuckle axis of the Packard axle is set at an angle of 10 deg. from the vertical; theoretically, it intersects the point where the tire contacts with the ground. With the recommended brake adjustment, it is very difficult to lock any of the wheels when driving upon dry pavement. It is possible, of course, and in this case the rear wheels will probably lock first. Mr. Vincent recommended that four-wheel brakes not be made so sensitive that they can be locked easily.

The involute form of tooth is used in the Packard transmission gears, the pressure angle being in the neighborhood of 18 deg. The gears are made of chrome-nickel steel and are oil-treated.

SINGLE-EIGHT ENGINE COMPRESSION

The compression-pressure in the single-eight engine is approximately 75 to 80 lb. per sq. in., this being taken with the engine warm, the engine turning over at 120 r.p.m. and the throttle wide-open. The maximum engine-torque is 2730 lb-in. The complete car in proper adjustment will average between 12 and 14 miles per gal. of gasoline.

Eight-in-line engines are primarily a passenger-car development. Mr. Vincent did not believe that they would be adopted for airplane service. He did predict the passing of the eight-cylinder V-engine in airplane practice, the light high-powered 12-cylinder engine superseding it entirely in military work. It was his opinion that the 12-cylinder aircraft engine could be built with less weight per horsepower than any other type.

AMATEUR CABARET PLEASES

Cleveland's energetic and versatile Secretary, Leo Williams, spiced-up the dinner-hour with musical features. He introduced the Cleveland Four-in-Line, an organization of male choristers whose close harmony drew deserved applause. A young gentleman by the name of Ball contributed several tuneful accordion selections which pleased the assembly. Williams stepped from his secretarial rôle for a brief period and teamed with a young lady in a novel rendition of "That Red-Headed Gal." This naturally brought down the house. Coker F. Clarkson spoke briefly and presented a Scotch story gathered up during his recent European travels. Chairman John Younger presided at both the dinner and the meeting. Several members of the Council were seated at the guest table and enjoyed the meeting thoroughly.

WASHINGTON CANCELS OCTOBER MEETING

The Governing Committee of the Washington Section has sent the following announcement to all members of the Society in its territory, the Council and others interested:

> The meeting of the Washington Section of the Society of Automotive Engineers, scheduled for Oct. 5, has been cancelled as a mark of respect for the members of the Bureau of Standards who, as a result of the recent deplorable explosion, gave their lives in the pursuit of scientific advancement.
>
> Stephen M. Lee — Urban J. Cook
> James E. Kendig — Logan L. Lauer
>
> To the families of these men, the Section extends the deepest sympathies of its members. For those who were injured, it hopes for speedy recovery.
>
> Washington, D. C. THE GOVERNING COMMITTEE.
> Sept. 24, 1923.

HALE ON BALLOON TIRES AT INDIANA

At the Indiana Section meeting on Oct. 11, J. E. Hale, of the Firestone Tire & Rubber Co., will speak for low-pressure air-cushion tires, more generally known as "balloon" or "doughnut" tires, in spite of efforts to make the more dignified term current.

Indiana Section members and their guests who did not hear Mr. Hale at Spring Lake will have the opportunity to hear not only what he said then, but also additional material which has been worked out since the summer meeting. The film shown at Spring Lake will be a part of the program beginning at 8 o'clock, after an informal dinner at 6:30 at the Severin Hotel, Indianapolis.

HEAD-LIGHTING WITHOUT GLARE

R. N. Falge Discusses Possibilities of Its Accomplishment Before the Metropolitan Section

R. N. Falge

Activities in the Metropolitan Section were renewed on Thursday, Sept. 20, with the first regular monthly meeting of the fall season at the Automobile Club of America, New York City. Importance of Better Automobile Head-Lamps and Proper Adjustment was the topic for discussion. R. N. Falge and W. C. Brown, who read a paper[1] on this subject during the Semi-Annual Meeting of the Society at Spring Lake, elaborated their previous offering and illustrated their remarks with the aid of a stereopticon. The effects produced by placing the bulb at the focal point of the lamp, by moving it forward, backward, above and below this point along the reflector axis, and also by vertical flutes and horizontal prisms placed in the glass, were given ocular demonstration; and following this various lenses taken from the stocks of the best known manufacturers were placed before the lamp and their compliance with or variation from the standards determined for the most efficient illumination was noted. An active discussion followed, which was participated in by Dr. A. W. Devine, Illuminating Engineer of the State of Massachusetts, W. F. Little, of the Electrical Testing Laboratories of New York City, H. K. Maples of the American Automobile Association, and R. E. Carlson, of the Bureau of Standards, City of Washington; a communication also was read from F. H. Ford, of the C. A. Shaler Co. The discussion will be printed in an early issue of THE JOURNAL.

Mr. Falge said that the accurate construction and easy adjustability of automobile head-lamps, though they had received much attention, are still susceptible of further improvement; if motorists realized that the road would not only be illuminated better, but that objectionable glare could be avoided and a better and safer head-lamp secured by properly adjusting the head-lamps now in service, an immediate and far-reaching improvement would be effected. Adjustment consists first of focusing and then of directing the beam. Less than one-third the time now consumed would be required if the filaments were accurately placed by the manufacturer. Inaccuracies of adjustment of the filament are compensated for by placing a lens over the lamp, and by tilting. A type of lens was described that will compensate for commercial variations in the position of the filament and at the same time distribute the light effectively, converging or diverging the rays laterally while maintaining the top or cut-off substantially at the level of the head-lamp. As the maximum intensity should be placed as near to the top as possible in order to produce the best illumination of the roadbed, and as the eye accommodates itself slowly to changes of intensity, the lens takes advantage of these characteristics by spreading the light in each zone by different amounts.

SPOT-LIGHTS

Mr. Falge added that in driving to the Summer Meeting at Spring Lake his car was equipped with two spot-lights in addition to head-lamps and that the spot-lights were turned off when other cars approached. Dr. Devine remarked that spot-lights could not be used in Massachusetts.

NEEDED IMPROVEMENTS

Dr. Devine further stated that greater accuracy in the production of reflectors is very much needed. He felt that with that accomplished and the elimination of necessity for adjustment of the bulb in the lamp, all that would be necessary in the proper installation of head-lamps would be to tilt them a proper amount. He pointed out that it is essential that in aiming the head-lamps the rear passenger load or its equivalent be in place. He referred to the fact that 11 Eastern States have been proceeding in conference with the purpose of so harmonizing conflicting laws of bordering States that they will not cause inconvenience to motorists.

CONDITIONS IN MASSACHUSETTS

Stating that the head-lamp adjustment is the most difficult thing to attain, the relatively severe enforcement of head-lamp laws in Massachusetts was taken up. There are 12 inspectors in the field there now and 12,000 complaints are in hand. Last year the number of fatal accidents during the night time was reduced 50 per cent by legal enforcement and regulation, but the headway made was not permanent because during two months this summer fatalities increased 50 per cent as compared with last year. This has resulted in the institution of another campaign of regulation and examinations of head-lamps are being made at the rate of 20,000 per month.

One further point that Dr. Devine made was that the reflectors must be maintained in good condition to secure good illumination, the reflectors becoming discolored if damp for a short time. In Massachusetts the motorists are given sketches showing a cross-section of properly and of improperly focused head-lamp illumination. They are told to stand their cars 25 ft. from a board and secure the desired focus and then tilt the lamps down the required amount. Eight hundred different stations in Massachusetts have been authorized by the State to adjust head-lamps.

THE MOTORIST'S POSITION

Mr. Maples asserted that lamps with different kinds of adjustment would not stay in adjustment because of the poor quality of either the lamps or the sockets; that not the motorists but the manufacturers are to blame; that the principal problem is to determine standards for the lamps and for the sockets.

WORK OF BUREAU OF STANDARDS

Mr. Carlson stated that the Bureau of Standards and the Illuminating Engineering Society had cooperated in the preparation of specifications and that 27 different head-lighting devices had been accepted as satisfactory, but that these are of no value except when properly adjusted. After testing 400 cars only 5½ per cent were found to conform to requirements. Letters from 50 automobile manufacturers showed great diversity of opinion as to what should be done. Head-lamps may be thrown out of adjustment even before the cars are delivered to customers.

Mr. Carlson urged that the matter of securing proper adjustment have the hearty cooperation of all concerned. On behalf of the Bureau of Standards, he is now working through local trade associations and dealers. Screens for use in securing proper adjustment, like those installed by

[1] See THE JOURNAL, July, 1923, p. 25.

the Bureau in the District of Columbia, are furnished. He felt that the Society should take a very active part in this matter, through its Sections particularly, and that proper adjustment should be available to all at proper stations for a nominal fee. He advocated a system by which some member of the Society should be personally responsible for improvement of conditions in each part of the Country. The American Automobile Association and the National Motorists' Association have printed and distributed proper instructions. Safety Councils and those charged with the enforcement of the law should naturally take part in the proceedings.

Mr. Ford suggested that it would assist the user if illumination were expressed in such terms as "bright moonlight," instead of candlepower, focal distance, and the like. He said that most instructions for adjusting head-lamps are not intelligible to the motorist and that the latter does not comprehend what he is trying to get, and consequently cannot get it.

Replying to questions, Mr. Falge said that the Society is now endeavoring to bring about practice in the construction of head-lamps that will prevent the lenses from turning from their proper position for satisfactory illumination; that "fixed" adjustments of bulbs could not be depended on entirely because a very slight change in adjustment produces a large variation in the distribution of light; that it would be possible to compel motorists to keep the head-lamps in adjustment if they knew they would be penalized for not doing so; that in Massachusetts the proportion of complaints was 3 for insufficient light to 1 on account of the lens. Adjusters are employed in Massachusetts who notify drivers to adjust certain parts and give them cards that must be o.k'd. Reports show that 36 per cent of the head-lamps inspected had twisted lenses, 73 per cent were out of focus, and 70 per cent had tilted reflectors.

Chairman Myers, in summing up the discussion, concluded that the two important points to be considered are (*a*) giving the users in general some simple method of checking-up in order to get proper adjustment, and (*b*) securing greater refinement in the manufacture and assembly of head-lamp equipment to preserve alignment and focus.

Procedure for Head-Lamp Adjustment

Supplementing his original paper, Mr. Falge submitted the following suggestions for adjusting head-lamps:

(1) *Use Only 21-cp. Gas-Filled Head-Lamp Bulbs.*—Lamp-bulbs should be uniform and efficient in performance throughout their life. They should have highly concentrated filaments to give the minimum beam divergence. The filaments should be placed accurately with respect to the locking-pins and to the axis of the base in order to minimize beam distortion. They should not sag in service

If new lamps do not give a fairly white light with the engine running at a speed corresponding to 18 or 20 m.p.h., have the battery inspected at a service-station before proceeding with the adjustment. If lamps have been burning-out at short intervals, have the connections from the generator to the battery inspected, and replace any socket, fuse or connection that is found to be defective

(2) *Polish Reflectors with a Soft Cloth Dipped in Powdered Lamp-Black.*—Fogged, rusted or defective reflectors should be replaced with new ones; they should not be replated. If the gasket cord, which is provided to make an air-tight seal between the lens and the reflector, is loose, fasten it back in place with shellac

(3) *Focus the Lamps in the Reflectors.*—Some means, which can be determined upon examination, is always provided for moving the lamp forward and backward in the reflector. Adjust the lamp until the filament is so placed with respect to the reflector that the spot which is thrown on a vertical surface 25 ft. ahead of the car will be of the minimum size and approximately round. With fluted reflectors that spread the light to either side, the filament will be properly placed when the spot is shallowest in the vertical dimension. One point that should be kept in mind in either case is that the top of the beam should cut-off as sharply as possible, and results frequently can be improved by turning the lamp over in the socket when the beam is not satisfactory as first installed

(4) *Install Redirecting Equipment.*—When the ordinary redirecting lens is used, be sure that it is placed in the head-lamp door so that the wording on it reads properly from the outside and that it is fastened securely so that it cannot rotate away from a truly vertical position. When fluted reflectors are used, they in themselves accomplish the spreading of the light and require only a plain cover-glass. It is important that flutes be in a truly vertical position

Install one head-lamp door, with the glass properly fastened in it, on the head-lamp and hold the other door loosely in front of the other head-lamp in the proper position. Cover first one beam and then the other. The spots from both, as seen on the vertical surface, should look practically the same. If they are noticeably different, it means that the focus has been disturbed in installing the door, possibly due to the fact that the reflector moved back on the springs and the lamp position remained the same. Re-focus the lamp so that both beams will look the same. Install the other door and re-focus in the same manner

(5) *Aim the Beams.*—Place the car, with the rear seat fully loaded, on a level stretch with a

Schedule of Sections Meetings

OCTOBER

4—Detroit Section—Rotary Valves—E. Bournonville
Rotary Disc Valve—O. D. Heavenrich
11—Indiana Section—Low-pressure Air-cushion Tires—J. E. Hale
12—Mid-West Section—Fundamentals in Engineering—C. F. Kettering
18—Metropolitan Section—Brakes—H. M. Crane
23—New England Section—Duralumin—A Discussion of Some of Its Applications—R. W. Daniels
24—Milwaukee Members Meeting—City Planning in Relation to Traffic
29—Buffalo Section—Problems in the Construction of Low-Pressure Tires—J. F. Palmer
Cleveland Section—No meeting this month because of National Production Meeting in Cleveland, Oct. 25-27

garage door or other vertical surface 25 ft. ahead. Sight through the center of the rear window over the radiator cap and mark a point on the vertical surface at the same height as the lamp centers. Draw a horizontal line through this point and on it locate points at the right and left of the center line directly ahead of each head-lamp

On most makes of car a universal-joint or a slot under one of the bolts fastening the head-lamp to the fender or to the frame is provided to facilitate aiming. On a few cars, such as the Ford, it will be necessary to bend the head-lamp bracket. With one head-lamp covered, center the spot from the other on the vertical line through the point *A* directly ahead of it, and tilt the beam to the point where its top or cut-off is just below the horizontal line *AA*. In a similar manner the other head-lamp should be aimed on the point *A* directly ahead of it

When adjustments are made properly, a shallow band of light will be thrown upon the vertical surface with its top just below the horizontal but with a low intensity diffused light above the horizontal line. The road surface will be so well lighted that the motorist can pass other cars conveniently and safely. It is possible that oncoming drivers may at times signal to the motorist to dim. Intolerable glare has been so common in the past that whenever drivers see head-lamps illuminated with a white light they anticipate trouble

Under a few conditions glare results even with head-lamps properly adjusted as, for example, when a car is coming over a rise. Under such conditions, drivers should, of course, dim the head-lamps for both cars and pedestrians

(6) *Maintenance.*—If you can distinguish the faces of persons in approaching cars by the light from your own head-lamps, your lights are glaring decidedly. At reasonable intervals the adjustments should be checked by throwing the beam against a vertical surface or viewing it from about 100 ft. ahead of the car in order to be sure that vibration and handling have not changed the focus and aiming materially. The head-lamps should, of course, be re-focused and re-aimed each time a lamp is renewed

Proper head-lighting is impossible if the equipment is not kept in reasonably good condition. The time it takes to wash the *outer* surfaces of the lenses now and then is well spent. Each time a lamp is renewed, and oftener if necessary, the reflectors, the lamps and the inner surfaces of the lenses should be cleaned carefully

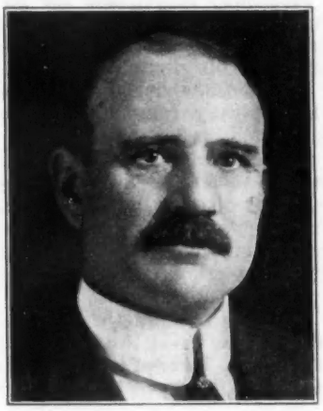
Metropolitan's next meeting will be held at the Automobile Club of America, 247 West Fifty-fourth Street, New York City, on Oct. 18. The topic is Brakes, and H. M. Crane is the speaker scheduled. The customary informal dinner will be served at 6:30 p. m. and the meeting will start at 8 o'clock.

DURALUMIN—SOME OF ITS APPLICATIONS

The popularity of plant inspection these Fall days has spread to the New England Section. The next meeting, scheduled for Oct. 23, will begin with an inspection of the Baush Machine Tool Co.'s plant in Springfield, Mass. After a dinner at 6:30 in the Highland Hotel, Springfield, the evening technical session will have as speaker R. W. Daniels of the Baush Machine Tool Co. His topic will be Duralumin —A Discussion of Some of Its Applications. With the factory visit earlier in the day, Mr. Daniel's paper will maintain the high standard of New England Section programs.

C. W. NASH MID-WEST SECTION HOST

President of Nash Co. Says Engineers Deserve More Credit for Benefits of Auto



C. W. Nash

Mid-West Section made an excellent start of the new Section year in its visit to the plant of the Nash Motors Co. at Kenosha, Wis., on Sept. 21. From the morning golf tournament, through the plant inspection personally conducted by C. W. Nash, to the dinner in the evening at which Mr. Nash made a short address, the Mid-West Section members felt that no pains had been spared to make their visit a red-letter occasion.

The laurels in golf for low gross score went to Harold Long, purchasing agent of the Nash company. H. J. Mellum, secretary of the company, won second prize for low net, and Walter J. Buettner, the Mid-West treasurer, carried off third prize. A special feature not on the program was a ball game between the Nash and Simmons teams which was won by the Nash nine, 2 to 1.

Mr. Nash met his guests at luncheon and remained with them for the rest of the day. They were unusually fortunate in having him personally conduct the inspection of the plant in the afternoon, for he has a remarkably intimate acquaintance not only with all his men, but also with all the operations of his factory.

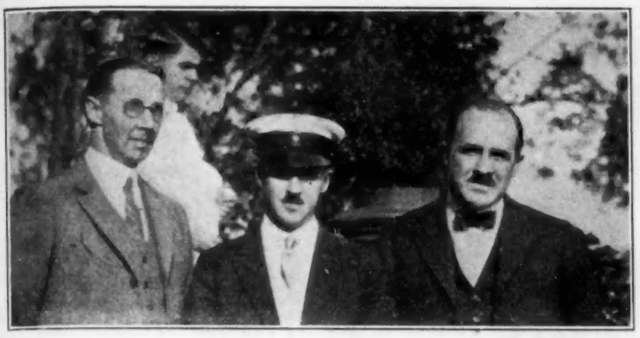
At the dinner which closed the visit, Mr. Nash expressed his belief in the engineering fraternity and his conviction that engineers should receive a larger share of the credit for the health and happiness which the motor car has brought to so many people. Mr. Nash attributes his success to close contact with his men and welcomes opportunities to reach engineers more intimately. The 50 Mid-West men present elected Mr. Nash the first honorary member of their Section, in appreciation of his expressed desire of securing better cooperation between the engineering and the production phases of the automotive industry.

The October meeting of the Mid-West Section will maintain the standard set by the Kenosha trip. C. F. Kettering, president of the General Motors Research Corporation, will speak on Fundamentals in Engineering at the meeting set for 8 o'clock on Oct. 12 at the Western Society of Engineers, Chicago, after the customary informal dinner at 6:30.

AIR-COOLING UNPOPULAR IN DETROIT

Section Opens Season Auspiciously Despite Frigidity of Cooling Mediums

Air-cooling and water-cooling have their proper places in motor vehicle design, but when they start on a rampage and attempt to make merry amidst a sport outing of the engineering fraternity, that is quite another matter. Detroit Section officers had planned an attractive program of swimming and boating events as a feature of the Section outing held on Sept. 10 at the Detroit Boat Club. On the particular afternoon under discussion, the air and water temperatures reached a point on the Fahrenheit scale that is far from conducive to comfort, especially when one is immersed in the colorful waters of the Detroit River. Just to prove the courage of the committee which had conceived the marine program, George McCain, its chairman, splashed about in hopes of selling the idea to others in the assembly, but without appreciable success. R. H. Weinert bolstered up the



H. M. Rugg C. H. Brennan G. L. McCain
Responsible for Detroit Section's Social Meeting

nautical program by cutting capers in his amazingly active motor boat "S. A. E." He found many willing riders in the assembly, particularly those of feminine gender. If maneuverability counts in supplying thrills to motorboat followers, the "S. A. E." takes top rank.

Temperatures bordering on frigidity had a beneficial effect on the other features of the program and the net result was a very successful social meeting. Over one hundred were seated at the dinner table and partook of the Boat Club chef's handiwork. This dusky individual soon announced that "the cupboard was bare" and saxaphone bleatings enticed the members and their ladies into the twirling of merry steps upon the ballroom floor. Acquaintance and congeniality were promoted by the injection of several Paul Jones and dancing continued for several hours. The number of ladies who expressed curiosity concerning the scheduling of a similar occasion in the near future serves as supporting evidence that the Detroit Section season was opened auspiciously.

Next Detroit Section Meeting

Various types of rotary and sleeve-valve engines will be described and discussed at the next meeting of the Detroit Section Thursday evening, Oct. 4. E. Bournonville will read a paper on valves of the rotary-tube type and Osmond D. Heavenrich will present the advantages of the rotary-disc valve. Well-known authorities on sleeve-valve engines are expected to attend and give short talks on the latest developments in this type. Of course the poppet-valve supporters will be on hand to defend the conventional practice now followed by a majority of the car producers. The combination of these several viewpoints on one program should produce an active and interesting discussion, to say the least. Detroit Section meetings are held in the General Motors Building and are preceded by an informal supper at 6:30. Presentation of papers starts promptly at 8 o'clock.

LOW-PRESSURE TIRE CONSTRUCTION

J. F. Palmer of the Hewitt Rubber Co. will speak on Low-pressure Tire Construction at the Buffalo Section meeting on Oct. 29. Mr. Palmer's experience—one of the widest in the industry—qualifies him to speak with authority on tire matters. The meeting, which will begin promptly at 8 o'clock, will be held in the Statler Hotel.

ANNUAL MEETING PAPERS WANTED

Members who are planning to present papers on the program of the Annual Meeting in Detroit next January should communicate with the New York office of the Society without delay. It is planned to make this meeting one of the most comprehensive ever held by the Society. There will be sessions on nearly every phase of automotive engineering; fuel, engines, chassis, bodies, production, aeronautics and other important topics will be treated.

The Meetings Committee is anxious to receive papers on any subject of general interest for consideration as Annual Meeting contributions. In submitting requests for a place on the program, give a reasonably complete description of what you intend to present. But, most important point of all, acquaint the Committee with your desire to appear on the program without waiting for an invitation.

WHEEL ALIGNMENT DISCUSSED

Successful Opening Session of New England Section

John F. Duby

Over 60 members and guests were present at the session of the New England Section of the Society held in Boston at the Buckminster Hotel on Sept. 11. Chairman R. E. Northway told of the plans for the season, the next meeting being scheduled to be held in Springfield on Oct. 2.

The September meeting was preceded by a dinner at which the best of good fellowship prevailed.

John F. Duby of Boston gave a valuable analysis of the features to be considered in wheel alignment and the troubles attributable to misalignment. He said that he had in his long experience found considerable confusion on the subject among engineers. Mr. Duby's purpose was to give a short, easy method of calculating and allowing the correct setting for all cars to assure good steering and minimum tire wear. He stated that it was not necessary to sacrifice the tire to secure good steering. Wheel alignment is seriously affected by axles, tie-rods, king-pins, radius-rods, steering-arms, steering-gear, and spring suspension plays an important part.

The rear wheels should be parallel in every way; that is, each wheel should revolve in a perfectly true circle. Some causes for misalignment of the rear wheels are bent axle-housing, improperly adjusted radius-rods and loose spring-clips.

There are very few cars on which the front wheels are the same distance apart at their centers as the rear wheels, and very few, if any, cars on which the front wheels are parallel to each other. They are closer together at the bottom than at the top, and the front axle is set at an angle to cause the king-pins to be further forward at the bottom than at the top. It is generally agreed that these conditions are correct; the disagreement arises over the reasons and the proper amount to be allowed.

The propulsion of the car tends to cause the front wheels to toe-out. This is one of the principal reasons for toeing them in. The purpose of camber is to bring the contact of the front tires on the ground directly under the king-pins. Also, the camber counteracts the effect of the weight on the spindles. As the camber increases, the tendency of the wheels to toe-out decreases. Large wheels require more toe-in than small ones. The author expressed the opinion that the proper amount of toe-in is from ⅛ to ¼ in., measured between the tires at points 9 in. from the ground. Also that cars will steer well with from 1 to 3 deg. of camber, but that there is undue tire wear with more than 2 deg. The object of tilting the front axle backward is to secure caster effect on the wheels, the tire contact with the ground being behind the line drawn through the king-pin axes. Insufficient tilt causes the wheels to pull hard in turning corners with the car. The front wheels should be the same distance apart as the rear wheels, at the bottom.

Wheel alignment should be measured after mounting bodies on cars, to assure proper axle tilt.

A lively discussion was had as to the effect of wheel alignment on tire wear and as to front wheel wabble.

Applicants *for* Membership

The applications for membership received between Aug. 15 and Sept. 14, 1923, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be send promptly.

ADAMS, MARTY MARTIN, stockroom supervisor, Diamond T Motor Car Co., *Chicago.*

BAKER, JAMES T., instructor, School of Automotive Electricity, *Milwaukee.*

BARNES, H. H., assistant maintenance engineer, Durant Motor Co. of Michigan, *Lansing, Mich.*

BLEIWEISS, M., branch manager, Templar Motor Co., *Cleveland.*

BRUCE, ORVILLE O., designer, Falls Motor Corporation, *Sheboygan Falls, Wis.*

CHILDS, KENNETH L., research engineer, Standard Textile Products Co., *New York City.*

COSHOF, MORRIE E., student, Tri-State College of Engineering, *Angola, Ind.*

DROGITCH, M. G., draftsman, Durant Motors, Inc., *Long Island City, N. Y.*

DUDLEY, J. KYLE, sales engineer, E. F. Houghton & Co., *Philadelphia.*

DUNNING, HUGH C., assistant general manager, Olds Motor Works, *Lansing, Mich.*

EDGARTON, LEWIS S., instructor, Mechanical Institute, *Rochester, N. Y.*

GRABFIELD, J. J., student, Massachusetts Institute of Technology, *Cambridge, Mass.*

GRIDLEY, W. T., vice-president and general manager, Depot Motor Bus Lines, Inc., *Chicago.*

GROETZINGER, C. A., engineer, Timken Roller Bearing Co., *Canton, Ohio.*

GUSTAFSON, WALTER A., draftsman, New England Telephone & Telegraph Co., *Quincy, Mass.*

HODGE, ROY C., purchasing agent, Cleveland Automobile Co., *Cleveland.*

JOHNSON, HOMER E., assistant engineer, Simms Magneto Co., *East Orange, N. J.*

JONES, WALTER R., aeronautical engineer, Boeing Airplane Co., *Seattle, Wash.*

KNOX, HARRY A., chief automotive engineer, Rock Island Arsenal, *Rock Island, Ill.*

KRANZER, EMANUEL C., designing engineer, Thomas E. Murray, Inc., *New York City.*

KUEHNE, OSCAR, student, Tri-State College of Engineering, *Angola, Ind.*

LEINDORF, EMIL, president and general manager, Leindorf Motor Sales Corporation, *New York City.*

LOEFFLER, BRUNO, designer, International Motor Co., *New York City.*

LUMM, NEWTON, experimental foreman, McCook Field, *Dayton, Ohio.*

POWELL, ROBERT E., consulting motor-transportation engineer, *Onancock, Va.*

RICHARDSON, EDWARD ADAMS, weight controller, Curtiss Aeroplane & Motor Co., Inc., *Garden City, N. Y.*

ROANTREE, T. C., mechanical engineer, Westinghouse Electric & Mfg. Co., *East Pittsburgh, Pa.*

Applicants *Qualified*

The following applicants have qualified for admission to the Society between Aug. 10 and Sept. 10, 1923. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

ADAMS, A. STUART (E S) student, University of Michigan, Ann Arbor, Mich., (mail) 2239 Hughes Terrace, *Detroit.*

AITKEN, LEON L. (J) chief tool designer, Fuller & Sons Mfg. Co., Kalamazoo, Mich., (mail) 317 Perry Court, *Flint, Mich.*

ARGELAN, PETER ARTHUR (E S) student, University of Cincinnati, *Cincinnati,* (mail) 2358 Clifton Avenue.

BECK, MILTON (M) chief engineer, Stanley Motor Carriage Co., *Newton, Mass.,* (mail) 196 Tremont Street.

DONALDSON, W. R. (J) designer, General Motors Truck Co., *Pontiac, Mich.*

EIGE, ELMER H. (E S) student, University of Minnesota, Minneapolis, (mail) 221 North Cicero Avenue, *Chicago.*

GROSS, EDWARD H. (A) sales manager, Donning Carburetor Corporation, Bridgeport, Conn., (mail) 3176 Lenox Avenue, *Detroit.*

GUGLER, JULIUS H. (A) vice-president, Globe Electric Co., *Milwaukee,* (mail) 231 Martin Street.

HAMPTON, E. H. (A) sales engineer, Hampton-Watson & Cia, Cerrito 702, *Buenos Aires, Argentina.*

HARRINGTON, GEORGE R. (M) assistant engineer, United Motors Products Co., *Grand Rapids, Mich.,* (mail) 412 Crescent Street, Northeast.

HORI, HISASHI (F M) chief engineer, Yanese Automobile Co., Ltd., 2, Jenigqmecho, Kojimachiku, *Tokyo, Japan.*

LANDES, JAMES U. (A) engineer, Standard Oil Co., *Portland, Ore.,* (mail) 149 Grand Avenue.

MACK, WILLIAM (A) sales manager, Borg & Beck Co., 920 South Michigan Avenue, *Chicago.*

MICHELIN, MARCEL (F M) manufacturer, Michelin & Cie, *Clermont-Ferrand, France.*

PATON, CLYDE R. (A) teaching assistant in mechanical engineering, University of Michigan, *Ann Arbor, Mich.,* (mail) 1121 Church Street.

POLLITT, CAPT. ALLEN W. (S M) shop superintendent, Quartermaster Corps, *Camp Normoyle, Tex.*

PORTER, R. C. (A) engineer, Stromberg Motor Devices Co., 250 West 57th Street, *New York City.*

RAWLINS, EDWARD L. (A) sales engineer, Arthur L. Eaton Co., 3769 Moneta Avenue, *Los Angeles.*

RIES, BEN A. (M) chief engineer, truck equipment division, Edward R. Bacon Co., 735 Folsom Street, *San Francisco.*

SAUNDERS, LAURENCE P. (M) experimental engineer, Harrison Radiator Corporation, Lockport, N. Y., (mail) 10-126 General Motors Building, *Detroit.*

SHARLOCK, HERBERT L. (A) vice-president and director of sales, Gernandt Motor Corporation, *South Bend, Ind.*